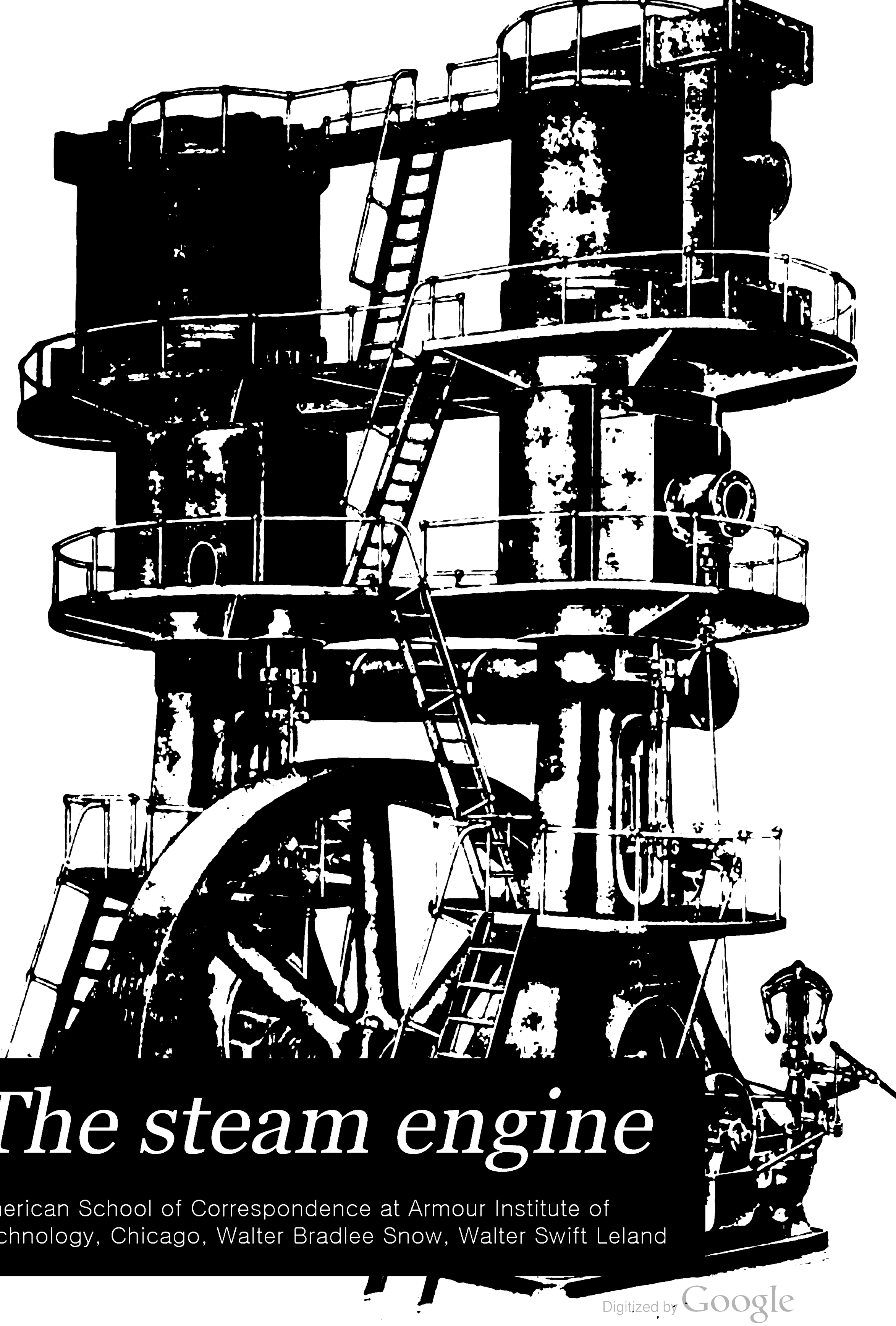
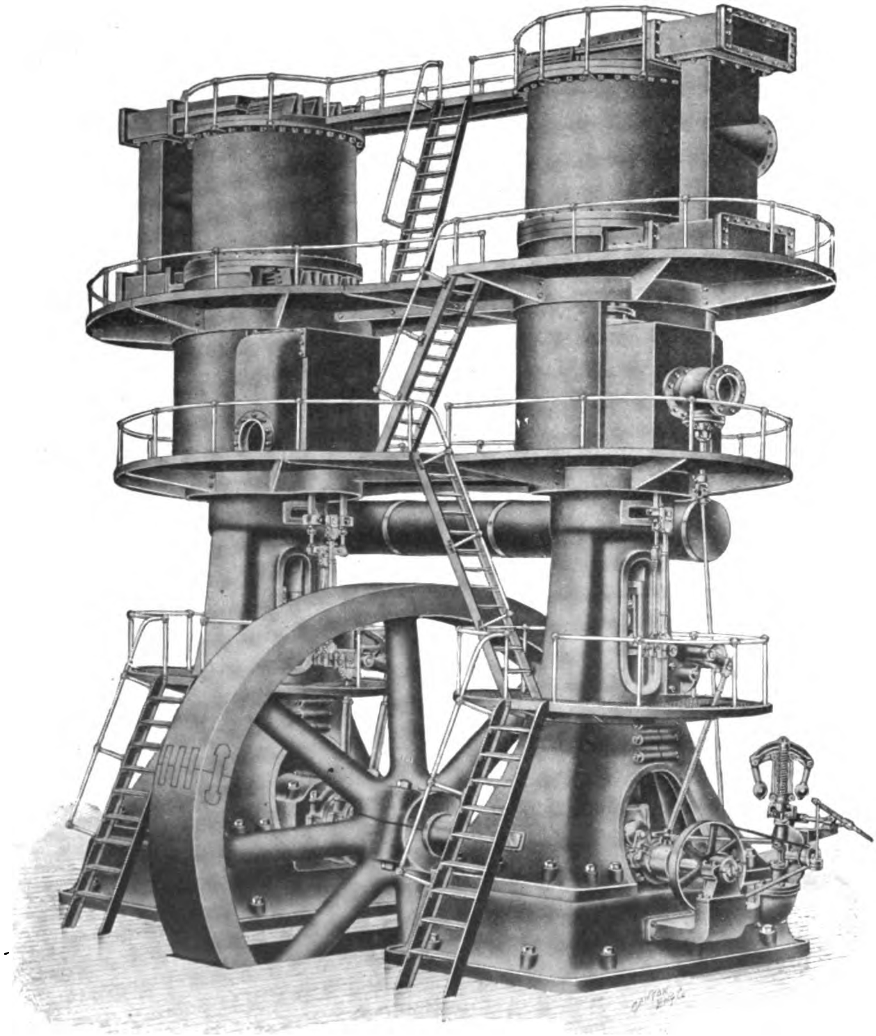


The steam engine

American School of Correspondence at Armour Institute of Technology, Chicago, Walter Bradlee Snow, Walter Swift Leland



VERTICAL CROSS COMPOUND BLOWING ENGINE.
Buckeye Engine Company.

The Steam Engine

A Practical Guide to the

CONSTRUCTION, OPERATION, AND CARE OF STEAM ENGINES, STEAM TURBINES, AND THEIR ACCESSORIES

THE STEAM ENGINE—PART I

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THE STEAM ENGINE—PART II

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ILLUSTRATED

CHICAGO
AMERICAN SCHOOL OF CORRESPONDENCE
1908

T J A
K E

Foreword

IN recent years, such marvelous advances have been made in the engineering and scientific fields, and so rapid has been the evolution of mechanical and constructive processes and methods, that a distinct need has been created for a series of *practical working guides*, of convenient size and low cost, embodying the accumulated results of experience and the most approved modern practice along a great variety of lines. To fill this acknowledged need, is the special purpose of the series of handbooks to which this volume belongs.

¶ In the preparation of this series, it has been the aim of the publishers to lay special stress on the *practical* side of each subject, as distinguished from mere theoretical or academic discussion. Each volume is written by a well-known expert of acknowledged authority in his special line, and is based on a most careful study of practical needs and up-to-date methods as developed under the conditions of actual practice in the field, the shop, the mill, the power house, the drafting room, the engine room, etc.

¶ These volumes are especially adapted for purposes of self-instruction and home study. The utmost care has been used to bring the treatment of each subject within the range of the com-

mon understanding, so that the work will appeal not only to the technically trained expert, but also to the beginner and the self-taught practical man who wishes to keep abreast of modern progress. The language is simple and clear; heavy technical terms and the formulæ of the higher mathematics have been avoided, yet without sacrificing any of the requirements of practical instruction; the arrangement of matter is such as to carry the reader along by easy steps to complete mastery of each subject; frequent examples for practice are given, to enable the reader to test his knowledge and make it a permanent possession; and the illustrations are selected with the greatest care to supplement and make clear the references in the text.

¶ The method adopted in the preparation of these volumes is that which the American School of Correspondence has developed and employed so successfully for many years. It is not an experiment, but has stood the severest of all tests—that of practical use—which has demonstrated it to be the best method yet devised for the education of the busy working man.

¶ For purposes of ready reference and timely information when needed, it is believed that this series of handbooks will be found to meet every requirement.

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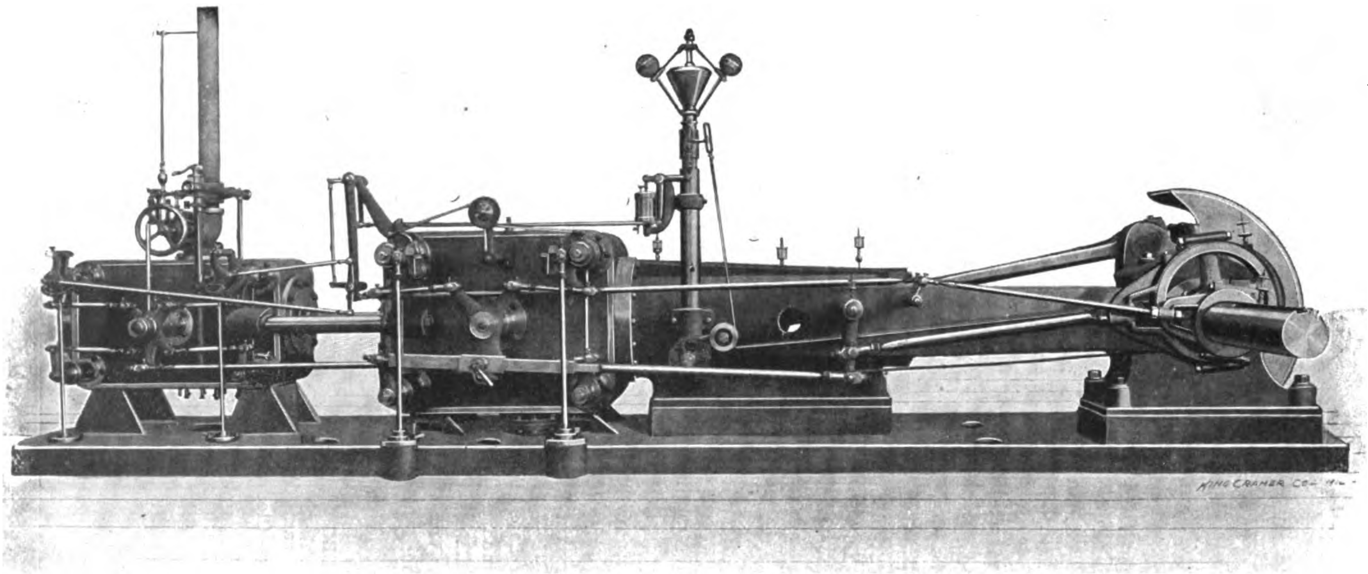


TANDEM CORLISS ENGINE DRIVING ELECTRIC LIGHT PLANT. TRADESMEN'S NATIONAL BANK, PITTSBURG.
Nordberg Manufacturing Company.

THE STEAM ENGINE

PART I.

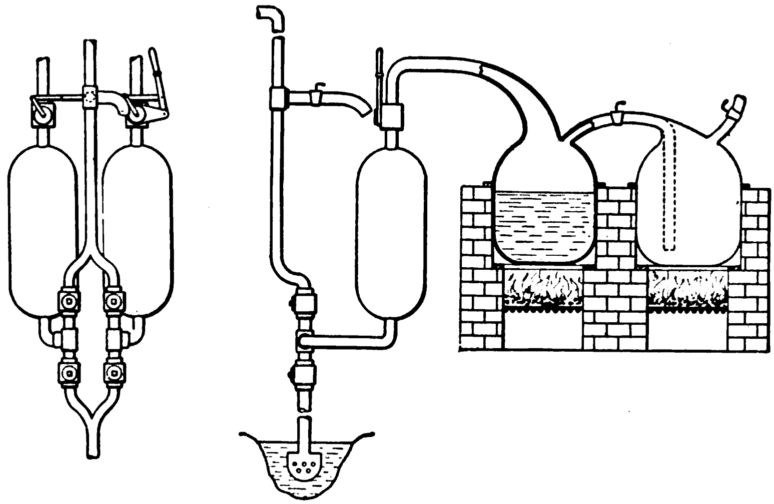
There are various kinds of engines from which mechanical work is obtained by the expenditure of heat. In the gas engine a mixture of gas and air is burned in the cylinder, the heat thus generated being converted into work by the expansion of the products of combustion. The action in oil and hot-air engines is very similar. The most important of all heat engines, however, is the steam engine, in which the heat in steam is transformed into work. It will be useful to review briefly some of the stages through which it has passed in its development.

The first steam engines of which we have any knowledge were described by Hero of Alexandria, in a book written two centuries before Christ. Some of them were very ingenious, but the best were little more than toys. From the time of Hero until the seventeenth century there was very little progress. At this time there began to be great need of steam pumps to remove water from the coal mines. In 1615, Salomon de Caus devised the following arrangement. A vessel, having a pipe leading from the bottom, was filled with water and then closed. Heat applied to the vessel caused steam to be formed, which forced the water through the pipe.

A little later an engine was constructed in the form of a steam turbine; but it was unsuccessful, and the attention of inventors was again turned to pumps.

Finally Thomas Savery completed, in 1693, the first *commercially successful* steam engine. It was very wasteful of steam as compared with our engines of today, but as being the first engine to accomplish its task it was a grand success. Savery's engine (Fig. 1) consisted of two oval vessels placed side by side and in communication with a boiler. The lower parts were connected by tubes fitted with suitable valves. Steam from the boiler was admitted to one of the vessels and the air driven out. The steam was then condensed and a vacuum formed by letting

water play over the surface of the vessel. When the valve opened, this vacuum drew water from below until the vessel was full. The valve was then closed and steam again admitted, so that on opening the second valve the water was forced out through the delivery pipe. The two vessels worked alternately. When one was filling with water, the other was open to the boiler and was being emptied. Of the two boilers, one supplied steam to the oval vessels and the other was used for feeding water to the first boiler. The second boiler was filled while cold and a fire lighted under it. It then acted like the vessel used by Salomon de Caus and forced a supply of feed water into the main boiler.



END VIEW. Fig. 1. SIDE VIEW.

A modification of Savery's engine, the pulsometer (Fig. 2), is still quite common. It is used in places where an ordinary pump could not be used and where extreme simplicity is of especial advantage. Its valves work automatically and it requires very little attention.

A serious difficulty with Savery's engine resulted from the fact that the height to which water could be raised was limited by the pressure which the vessels could bear. Where the mine was very deep it was necessary to use several engines, each one raising the water a part of the whole distance. The consumption of coal

in proportion to the work done was about twenty times as great as that of a good modern steam engine. This was largely, though not entirely, due to the immense amount of steam which was wasted by condensation when it came in contact with the water in the oval vessels.

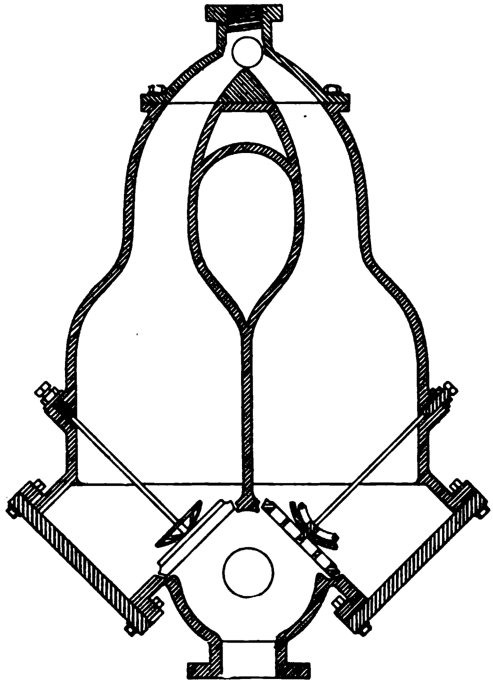


Fig. 2.

The next great step in the development of the steam engine was taken by Newcomen, who in 1705 succeeded in preventing contact between the steam and the water to be pumped, thus diminishing the amount of steam uselessly condensed. He introduced the first successful engine which used a piston working in a cylinder.

In Newcomen's engine, shown in Fig. 3, there was a horizontal lever pivoted at the center and carrying at one end a long heavy rod which connected with a pump in the mine below. A piston was hung from the other end of the lever, and worked up and down in a vertical cylinder, which was open at the top. Steam acted only on the lower side of the piston. Steam at atmospheric pressure was admitted from the boiler to the cylinder, and as the pressure was the same both above and below the piston, the falling of the heavy pump rod raised the piston. A jet of water was now passed into the cylinder to condense the steam and form a vacuum. This left the piston with atmospheric pressure above and very slight pressure below, so it was forced down and the pump rod again raised. Steam could again be admitted to the cylinder, the pump rod would fall, and so on indefinitely.

In the days of Newcomen it was very difficult to obtain good workmanship. For this reason it was often necessary to make the cylinders of wood, and even then there might be a space of one-eighth of an inch between the wall of the cylinder and the piston. In order to prevent steam from blowing through this passage, or air from leaking in when the steam was condensed, it was customary to keep a jet of water playing on the top of the piston.

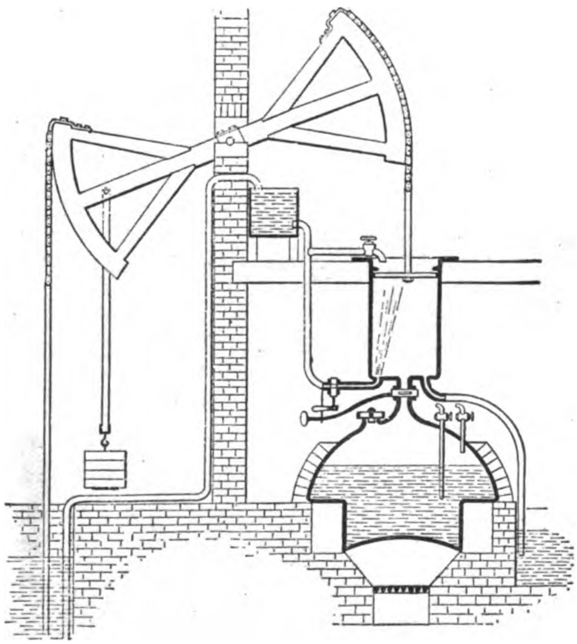


Fig. 3.

One great trouble with all these engines was that they required some one to open and close the cocks. Boys were generally employed to do this work. In order to get time to play, one of them rigged a catch at the end of a cord which was attached to the beam overhead. This did the work for him. Making the valves automatic in this way made it possible to dispense with the services of the boy and at the same time greatly increase the speed of the engine. This engine was improved slightly from time to time by different inventors and was very extensively used until

Watt's time. Some of them are in existence today. While this engine was a success and a great improvement over its predecessors, it was still very large, wasteful and heavy, in comparison with the work done. When the cylinders were made of iron they were simply cast and not bored, thus leaving a rough, stony coating over the iron, called the skin.

In the year 1763, a small model of a Newcomen engine was taken to the shop of an instrument maker in Glasgow, Scotland, to be repaired. This instrument maker, whose name was James Watt, had been studying steam engines for some time and he became very much interested in this model. He was a man of great genius, and before he died his inventions had made the steam engine so perfect a machine that there has been but one really great improvement in it since his time; namely, compound expansion. All other improvements have been merely following in the line of his suggestions and constructing what he could not for lack of good tools.

He found that to obtain the best results it was necessary, "*First, that the temperature of the cylinder should always be the same as that of the steam which entered it; and, secondly, that when the steam was condensed it should be cooled to as low a temperature as possible.*" All improvements in steam-engine efficiency have been in the direction of a more complete realization of these two conditions.

In order to keep the cylinder nearly as hot as the entering steam, Watt no longer injected water into the cylinder to condense the steam, but used a separate vessel or condenser. He made his piston tight by using greater care in construction, so that it was not necessary to have a water seal at the top. He then covered the top of the cylinder to prevent air from cooling the piston. When this was done he could use steam above as well as below the piston: this made the engine double acting.

Also, in the effort to keep the cylinder as hot as the entering steam, he enclosed the cylinder in a larger one and filled the space between with steam. This was not often done, however, and only of late years has the steam jacket been of much advantage. He also used steam expansively, that is, the admission of steam was stopped when the piston had made a part of the stroke; the rest

of the stroke was completed by the expansion of the steam already admitted. This plan is now used in all engines that are built for economy.

Other inventions made by Watt on his steam engine were: a parallel motion, that is, an arrangement of links connecting the

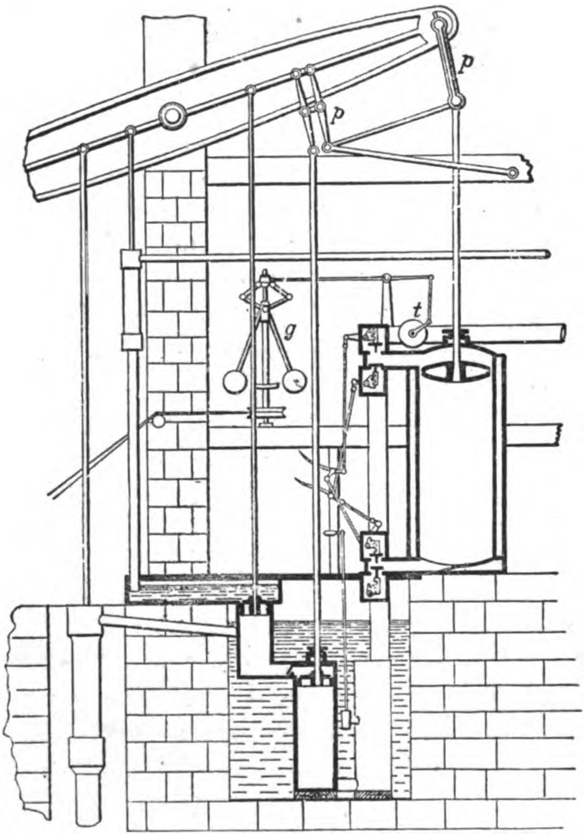


Fig. 4.

end of the piston rod with the beam of the engine in such a way as to guide the rod almost exactly in a straight line; the throttle valve, for regulating the rate of admission of steam and the centrifugal governor, which controlled the speed by acting on the throttle valve.

Watt also invented the "indicator," by means of which diagrams are made which show at all points the relation between the

pressure in the cylinder and the position of the piston at that instant. His assistant, Murdoch, invented the slide valve as a means of admitting and releasing the steam. Fig. 4 shows Watt's final engine.

Watt, like other early inventors, sold many of his engines to miners, who had been using horses to pump out the mines, and for this reason he rated his engines by the horse-power. Although this term has an historical derivation it has no real significance, and no relation whatever to the power of a horse. It is an established unit for measuring the rate at which work is done. One horse-power is the amount of work necessary to raise 33,000 pounds through one foot in one minute; and we may say that one horse-power is equal to 33,000 foot pounds per minute.

Watt saw that by using high-pressure steam he could get more work from it; but as it was not possible to make very reliable boilers he never used a pressure of more than seven pounds per square inch above the atmosphere. About the year 1800 comparatively high pressures came more into use and the *non-condensing* engine was introduced. In Watt's engine, and all those preceding his, a vacuum was produced in front of the piston by condensing the steam, and either the atmosphere or steam at atmospheric pressure pushed it through the stroke. In the non-condensing engine, using high-pressure steam, the space in front of the piston could be opened to the atmosphere at exhaust, and although the atmospheric pressure resisted its motion the pressure of the steam behind the piston was still greater than that of the air. These engines were much more simple than the condensing engines, as they required no condenser.

About this time what would now be called a compound engine was introduced by Hornblower and later by Woolf. It had two cylinders of different size. Steam was admitted into the smaller cylinder, and then passed over into the larger. The steam expanded a little in the smaller cylinder and much more in the larger one.

A great many attempts were made to build *locomotives*, but they were generally unsuccessful until George Stephenson built his engine, the "Rocket," in 1829. The principal new feature of this engine was the improved *steam blast* for increasing the

draft in the furnace and so making possible the use of a smaller boiler. Later he used the "link motion," which enabled the engine to be quickly reversed and the amount of expansion varied.

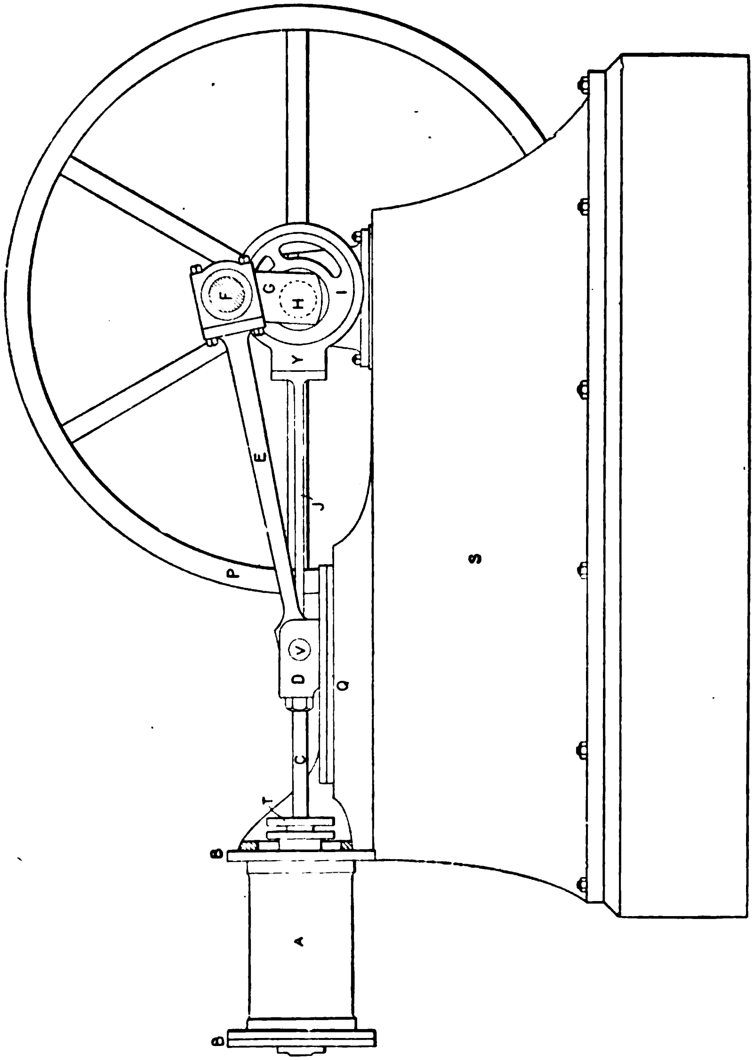


Fig. 5.

The Stephenson link motion may be seen on almost any locomotive. It is simply a device by which either of two eccentrics may be made to move the valve.

About the year 1814, Woolf introduced a compound pump-

ing engine in the mines of Cornwall, but a simpler engine was later introduced and Woolf's engine fell into disuse. This later engine became known as the Cornish Pumping Engine and was famous for many years because of its economy. It was the first engine ever built that could compare at all with modern engines in the matter of steam consumption. It consisted of a single cylinder placed under one end of a beam from the other end of which hung a heavy rod which operated a pump at the foot of the shaft. Steam was admitted on the upper side of the piston for a short portion of the stroke and allowed to expand for the remainder of the stroke. This forced the piston down, lifted the heavy pump rod and filled the pumps with water. Then communication was established between the upper and under side of the piston, exhaust occurred, and the heavy pump rod fell, lifting the piston and forcing the water out of the pumps. The cut-off was about .3 stroke, and the pump made about seven or eight complete strokes per minute with a short pause at the end of each stroke to allow the valves to close easily and the pumps to fill with water. The cylinder was jacketed. These engines needed great care and were in charge of competent men, to whom prizes were frequently given for the best efficiency, which doubtless accounts for their wonderful performance.

PARTS OF THE STEAM ENGINE.

Fig. 5 shows the elevation of a simple form of steam engine.

The Cylinder A (see plan, Fig. 6) is that part of the engine in which the piston moves back and forth. It is made of cast iron and accurately bored. Great care must be taken in this work, for any unevenness will allow steam to leak through between the piston and cylinder walls or it may even cause the piston to stick or work hard. In large engines the cylinder consists of two parts, the outer or cylinder proper and a comparatively thin cast iron liner. A space can be left between them for a steam jacket. Should the cylinder liner be damaged, it can be replaced without the expense of a new cylinder.

The Cylinder Heads B cover the ends of the cylinder and are securely bolted thereto. In the crank-end cover there is an opening for the piston rod to pass through. This opening is made

steam tight by a stuffing box which surrounds the piston rod. Sometimes the piston rod is prolonged beyond the piston and through the front cover. This extension of the piston rod is to

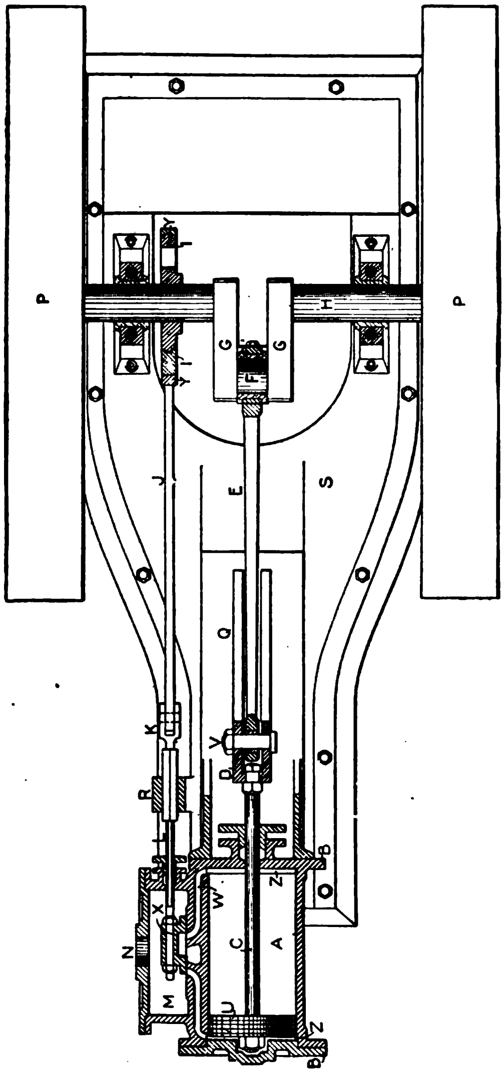


Fig. 6.

help steady the piston in a long stroke and is known as the tail rod. When a tail rod is used another stuffing box must also be provided for the head-end cover.

The Piston U, in small engines, is usually a thick disc of iron or steel, as shown in Fig. 6. It is often made conical, as shown in Fig. 7, to better withstand the steam pressure and to gain space. The piston must fit the cylinder steam tight and yet move easily. To accomplish this, one or more grooves in the piston are filled with packing (usually metallic), or spring rings may be used.

The Piston Rod C (Figs. 5 and 6) is made of steel and connects the crosshead and the piston to which it is rigidly fixed.

The Crosshead D serves to join the piston rod and connecting rod. At one end it is fastened to the piston rod, and at the other end is the wrist pin V on which the connecting rod swings. It is guided to and fro by the crosshead guides Q.

The Connecting Rod E is a steel forging from three to eight times the length of the crank, depending upon the type of engine. One end is jointed to the crosshead by the pin V, called the wrist pin, while the other encircles the crank pin and revolves with it. A detail view of one end is shown in Fig. 8; the other end is frequently similar. In some cases the small end is forked, as shown in Fig. 9.

The Crank Pin F forms the connection between the crank and connecting rod.

The Crank G, equal in length to one-half the stroke of the piston, converts the back and forth motion of the connecting rod into circular motion. It may be simply an arm, as shown in Fig. 10, or a complete disc keyed to one end of the shaft, as shown in Fig. 11. The disc is more nearly balanced than the crank.

The Shaft H transmits the rotary motion from the crank to the fly wheel P.

The Frame of the engine S is a heavy casting, which supports the cylinder and bearings. It should be securely bolted to the foundation.

The Steam Chest M receives steam directly from the boiler, and the steam passes thence through the ports W into the cylinder.

The Eccentric I is a disc keyed to the shaft so that its center and the center of the shaft do not coincide. The eccentric strap Y encircles the eccentric and imparts a reciprocating motion to the valve stem L and the eccentric rod J. This action is similar to

that of the crank and connecting rod, but exactly reversed. K is the valve stem crosshead and R its guides.

The Slide Valve X is the valve for alternately admitting

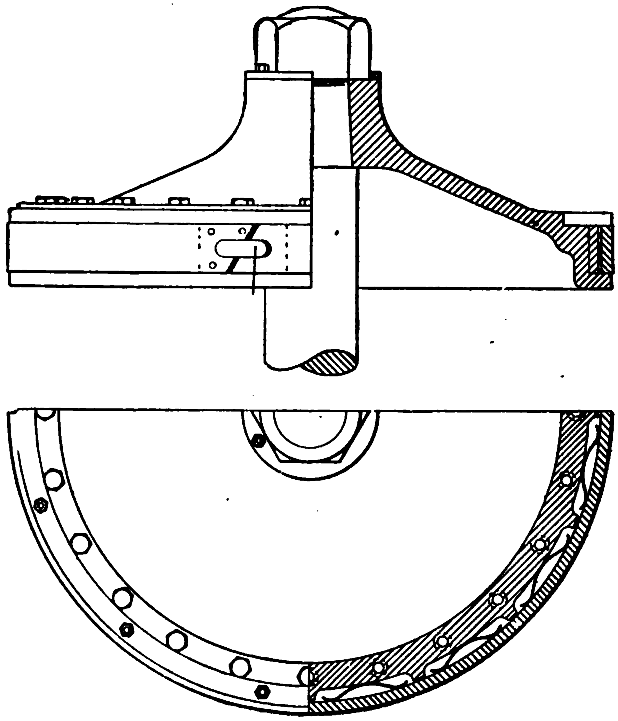


Fig. 7.

steam to the cylinder and releasing it. It has a cup-shaped cavity in its face through which the exhaust steam passes. It is situated in the steam chest and is moved by the valve gear, that is, the eccentric and the eccentric rod.

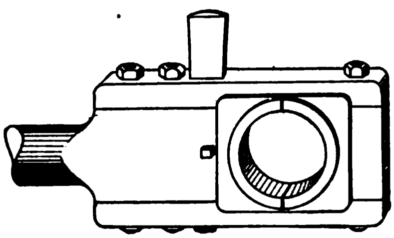


Fig. 8.

The Clearance Z is the space between the piston and the cylinder head (when the piston is at the end of the stroke), together with the volume of the steam ports. This volume must be filled with steam before the piston can start. It is usual to

express the clearance as a certain per cent of the volume swept through by the piston.

The crank may revolve in either direction. If we stand by the cylinder, facing the crank shaft, and the crank moves away from us as it passes over the shaft, we say that it is running *over*. If it moves away from us as it passes under the shaft, we say that the engine is running *under*. The action of steam in the cylinder of an engine is very complicated, and its discussion will be taken up later in the course.

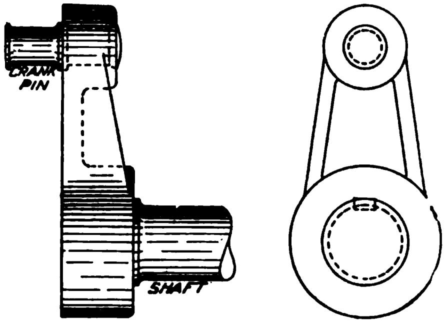


Fig. 10.

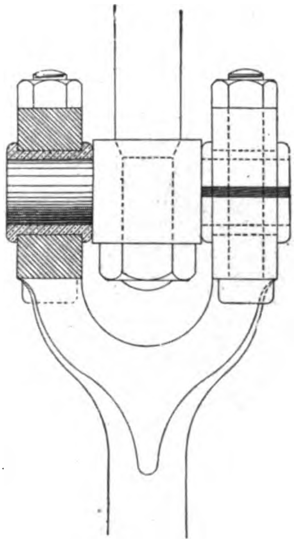


Fig. 9.

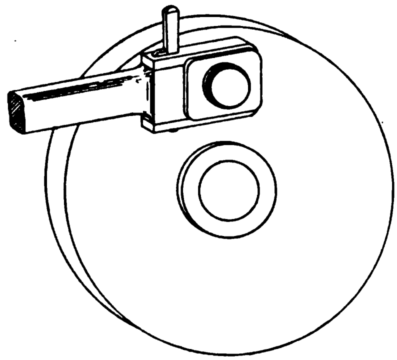


Fig. 11.

TYPES OF ENGINES.

Classification. There are so many different types of engines that it is difficult to classify them all properly. Most engines belong to several classes at one and the same time. For instance, there are condensing and non-condensing engines; there are

simple, compound, triple and quadruple expansion; there are high speed and low speed, vertical and horizontal, locomotive, stationary

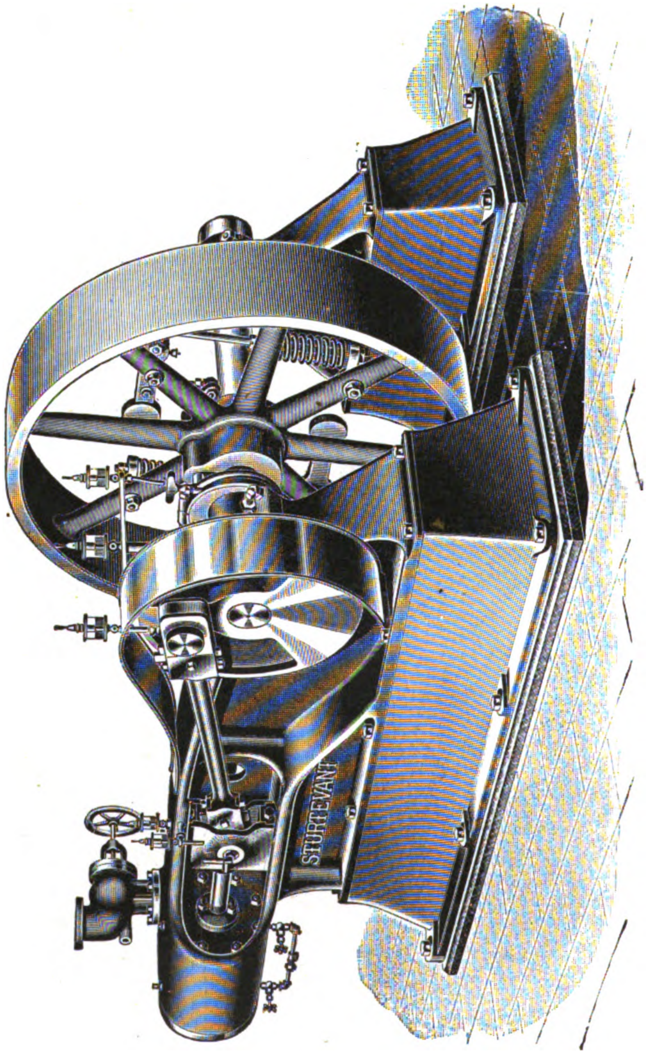


Fig. 12.

and marine, and many other classes into which these might be further subdivided.

Simple Engines. The simplest type of engine is the simple expansion. It has one cylinder and admits steam for a part of the stroke, expands it during the remainder and exhausts either into

the atmosphere or into a condenser. Simple engines (see Figs. 5 and 12) are now used only for comparatively small powers, say 100 H. P. or less, and although more extravagant of fuel than the others, may still be the most economical financially if low first cost is an important item, if they are not run continuously, or if the load fluctuates widely.

Compound Engines have two cylinders known as the high pressure and low pressure. Steam enters the smaller or high pressure cylinder and then expands until release, when it is exhausted into the larger cylinder, where the expansion is finished. The cylinders should be so proportioned that approximately the same amount of work can be done in each. The first cylinder is small, because it has the higher steam pressure, and a given weight of steam occupies less space when at high pressure. The second must be large, so that the volume at cut-off can contain all of the steam exhausted from the high.

Besides being more economical the compound has a distinct mechanical advantage. The two cranks may be set at right angles, so that when one is on dead center the other is at a position of nearly its greatest effort. This makes a dead center impossible, and gives a more uniform turning moment. Then the individual parts may be made lighter, and are thus more easily handled, but the engine is much more costly, and it is nearly twice as much work to take care of it.

When the cranks of a compound engine are at 90°, the low-pressure piston is not ready to receive steam when the high pressure exhausts, hence there must be a receiver to hold the steam until admission occurs in the low. Such engines are called *cross compound.* Fig. 13 shows one form. Sometimes instead of having the cranks at 90° they are placed together or opposite. Then the strokes begin and end together, and the high can exhaust directly into the low without a receiver. Such engines are called *Woolf engines.* A tandem compound engine, shown in Fig. 14, has both pistons on one rod, the high-pressure piston rod forming the low-pressure tail rod. Such engines are less expensive because there is but one set of reciprocating parts instead of two, but like simple engines they have the disadvantage of dead points.

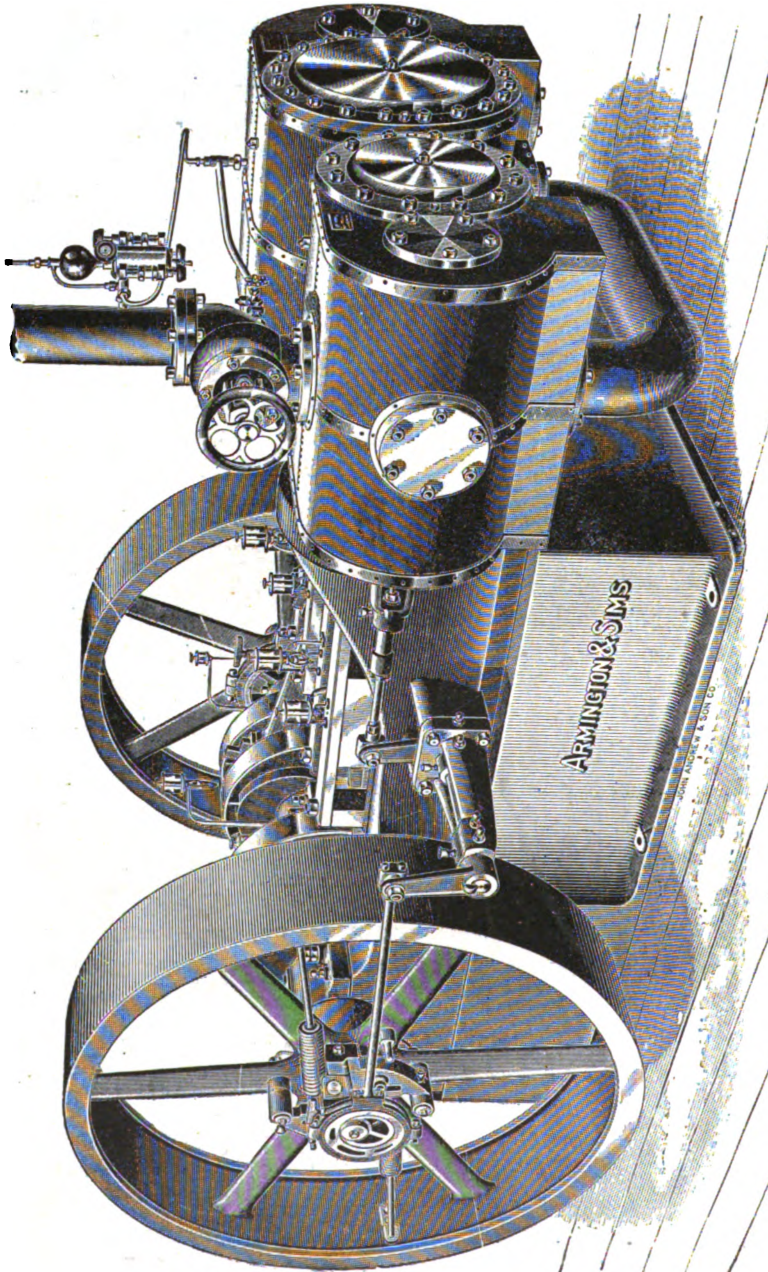
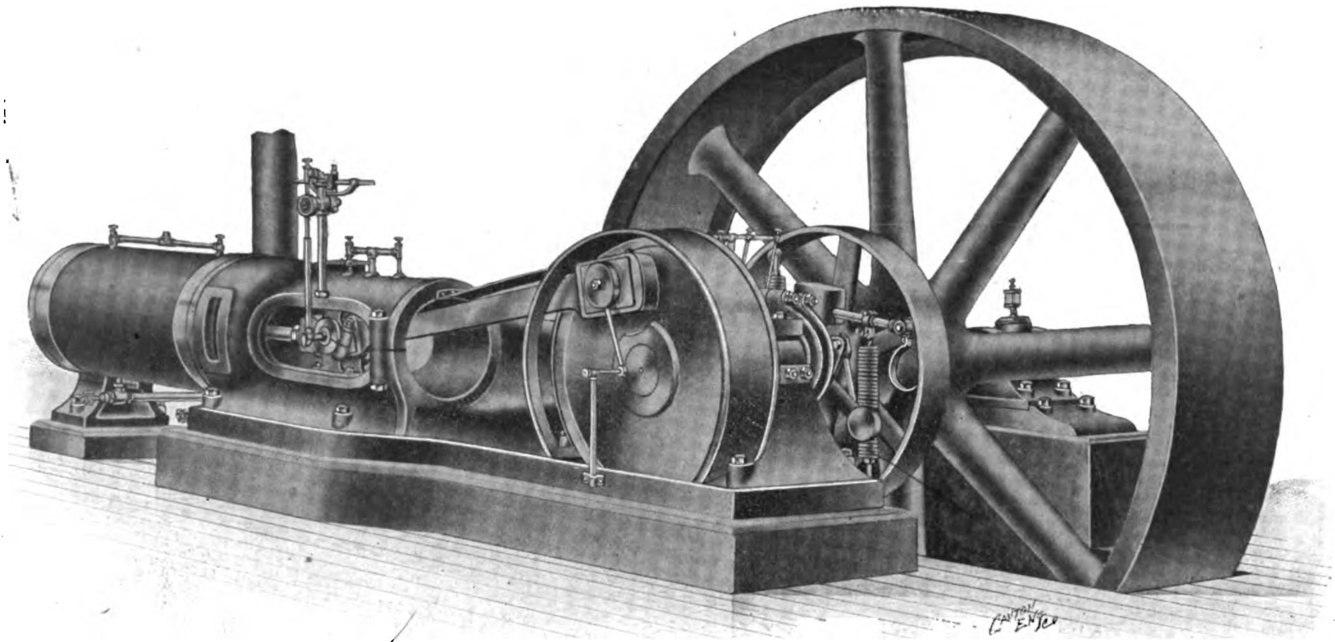


Fig. 13.



HEAVY DUTY, MEDIUM SPEED ENGINE.
Buckeye Engine Company.

Triple Expansion Engines expand the steam in three stages instead of two. There are usually three cylinders, the high, intermediate, and low, arranged with cranks 120° apart. This gives a more uniform turning moment than a compound. Sometimes there are four cylinders to the triple, one high, one intermediate, and two low. This arrangement gives better balance and is often used in marine work.

For triple engines there must be a receiver between each two cylinders. Fig. 15 shows the essential features of a triple expansion engine.

Quadruple Engines expand their steam in four stages instead of three. Multiple-expansion engines are nearly always condensing.

Cylinder Ratios. There are several considerations to be remembered when proportioning the cylinders of multiple-expansion engines. The ratio of the cylinders should be such that each develops nearly the same power; the drop in pressure between the cylinders and receivers should be small, and the strains in the cylinders about equal.

There are many formulas in use, some simple, others involving mathematical calculation. A common rule for compound engines is to make the ratio of the cylinders equal to the square root of the total ratio of expansion. Thus if the steam has a ratio of expansion of 9, the ratio of the cylinder volumes will be $\sqrt{9} = 3$, or the low-pressure cylinder will have a volume 3 times as great as the high-pressure cylinder. If the cylinder ratio is 3, and the length of stroke is the same for both, the diameter of the low-pressure cylinder will be 1.75 times that of the high-pressure cylinder.

Another rule is to make the cylinder ratio equal to the total ratio of expansion multiplied by the fractional part of the stroke completed when cut-off occurs in the high-pressure cylinder.

Suppose the ratio of expansion is 9, as above, and that cut-off occurs at $\frac{1}{3}$ of the stroke in the high-pressure cylinder. The ratio of cylinder volumes will be $9 \times \frac{1}{3} = 3$. If cut-off occurs at $\frac{1}{2}$ the stroke, the ratio will be $9 \times \frac{1}{2} = 4.5$.

For triple expansion engines the low pressure cylinder is made large enough to develop the whole power if steam at boiler pressure is used.

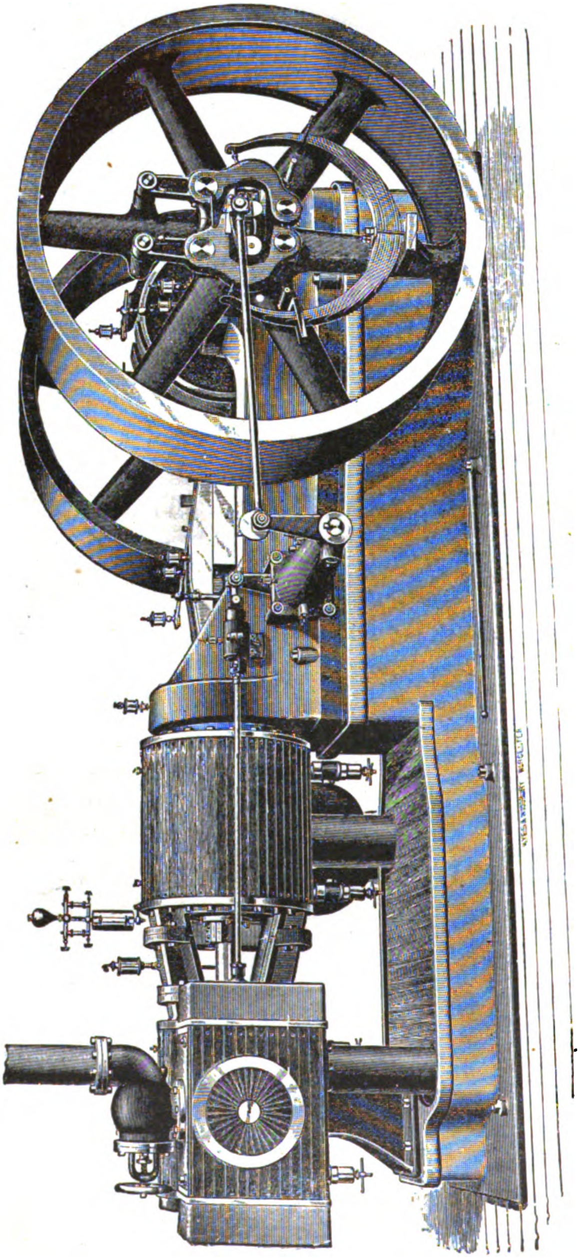


Fig. 14

The intermediate cylinder is made approximately of a mean size between the high and the low. The area of the intermediate piston is found by dividing the area of the low by 1.1 times the square root of the ratio of the low to the high.

We may write the above thus,

$$\text{Area of high-pres. cyl.} = \frac{\text{Area of low-pres. cyl.}}{\text{Cut-off of high-pres.} \times \text{ratio of exp.}}$$

$$\text{Area of inter. cyl.} = \frac{\text{Area of low-pressure cylinder.}}{1.1 \times \sqrt{\text{ratio of low to high.}}}$$

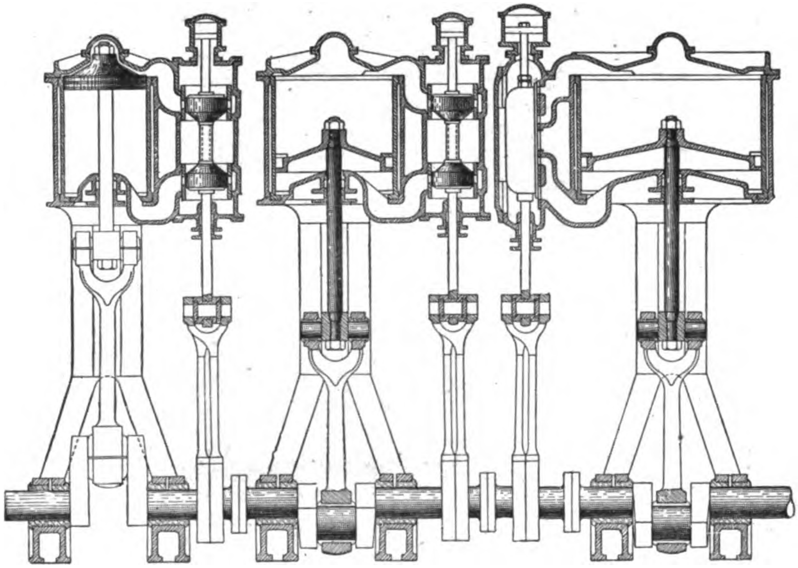


Fig. 15.

In general the volumes for triple expansion are of about the following ratios, 1 representing the volume of the high pressure cylinder:

1 : 2.25 to 2.75 : 5 to 8.

For quadruple-expansion engines the ratios are as follows:

1 : 2 to 2.33 : 4 to 5 : 7 to 12.

High Speed Engines. Of late years there has been a demand for engines of higher speeds than were formerly used. It was found desirable to run dynamo-electric machines by connecting them ***directly*** to the shaft of the engine rather than by belts as before.

This required engines running from 200 to 1,000 revolutions per minute instead of from 60 to 90 revolutions. Also for engines in

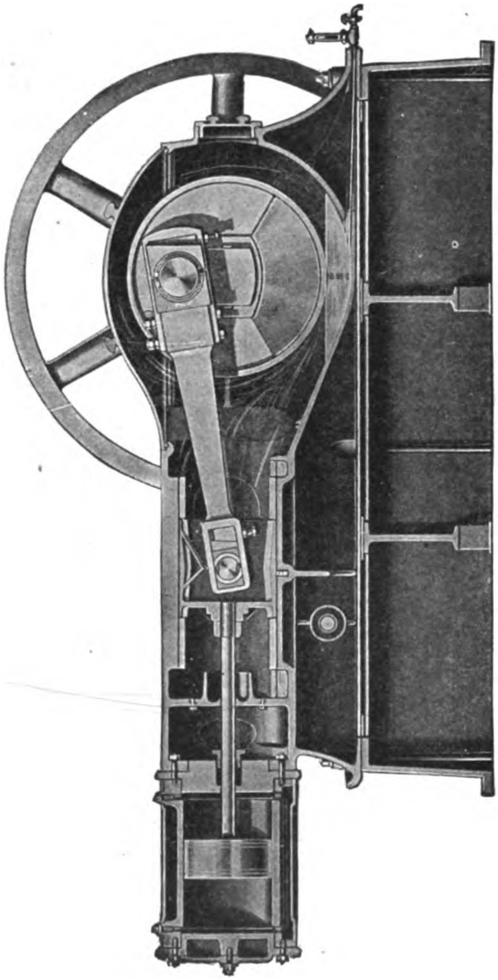


Fig. 16.

torpedo-boats, speeds as high as 400 or 500 revolutions are common.

Running at high speed requires various changes. Reciprocating parts must be made lighter to reduce the vibration, and must be

more carefully proportioned to maintain balance. Bearings must be made very much larger to reduce the pressure in order that the friction may not be excessive and cause heating. Special care is necessary that bearings should be tight, since the least looseness will cause knocking and hammering until the bearing is ruined. Parts must be as simple as possible and so arranged as to need the least possible attention. In slow-speed engines the engineer can watch the oil cups and oil any part while it is running. But this is impossible in high-speed engines, and special means must be used to insure plenty of oil to the bearings and the cylinder.

These peculiarities of high-speed engines may be easily seen in any engine for electric lighting, for running fans, etc. It is also necessary that the speed of engines for running electric generators should be very steady, as the slightest change in the speed make the lights flicker. The fly wheels are, therefore, larger or heavier than for other types and the governors are made specially sensitive. Fig. 16 shows a horizontal high-speed engine with the working parts encased. Fig. 17 shows a vertical high speed.

For any *double-acting* engine, that is, for any engine in which the steam acts first on one side and then on the other side of the piston, the piston first pushes and then pulls on the connecting rod and crank. At each half revolution of the crank the direction of the pressure reverses. It is this change of pressure which causes the pounding if the bearings are at all loose. This is one of the greatest troubles with high-speed engines. In order to avoid these rapid reversals in pressure, *single-acting* vertical engines are used to a considerable extent. In such engines the steam is admitted only to the *head end* of the cylinder. The other end stands open. The connecting rod is in compression throughout the whole revolution instead of being first in compression and then in tension. Besides insuring that the piston shall always push, this arrangement simplifies the valves.

There are several good single-acting, high-speed engines. One of the earliest made was introduced by Brotherhood. He used three cylinders, set around the shaft 120° apart. Another well-known example is the Willans "central valve" engine. These are both English engines. A well-known American engine

of this type is the Westinghouse high-speed engine, a section of which is shown in Fig. 18.

Vertical and Horizontal Engines. At the present time the most common type of engine is the horizontal direct-acting, that

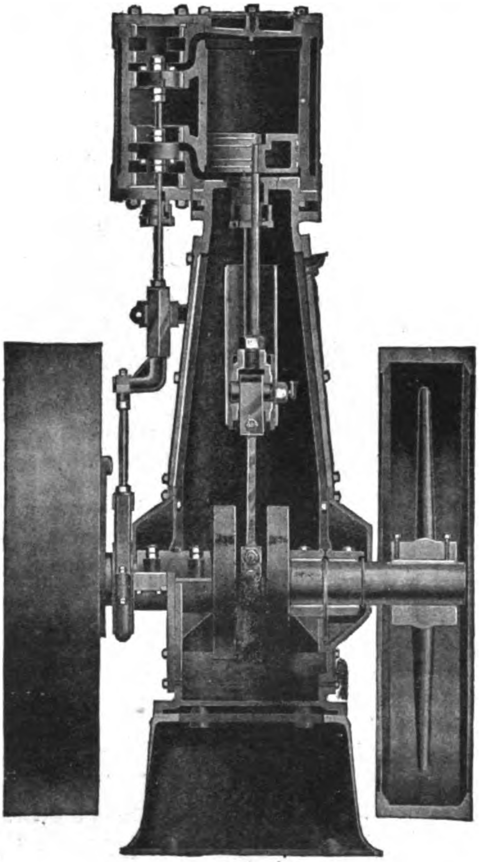


Fig. 17.

is, an engine whose cylinder is horizontal and whose piston acts on the crank through a piston rod and connecting rod. In small engines the whole is often on one bed plate. Such engines are called self contained. The cylinder is either bolted to the back of the bed plate or rests directly on it.

In marine work vertical engines are used in almost every case. The reason for this is, of course, the saving of floor space, which is so important in a vessel. This saving of space, however, is also

very important in many cases such as in crowded engine rooms of cities where land is expensive, and as there are a number of advantages which vertical engines have over horizontal, they are coming largely into use in stationary practice.

A second advantage of the vertical over the horizontal engine is the reduction of cylinder friction and unequal wear in the cylinder of the latter. In the horizontal engine the piston is generally supported by resting on the cylinder, which is gradually worn until it is no longer round. This causes leakage of steam from one side to the other. Evidently this is entirely avoided in the vertical engine.

Still another advantage of the vertical engine is the greater ease of balancing the moving parts so that there shall be no jarring or shaking. It is impossible to perfectly balance a steam engine of one or two cylinders. If it is balanced so that there is no tendency to shake sidewise, it will shake endwise; and if it is balanced endwise it will shake sidewise. The jarring is due to the back and forth motion of the reciprocating parts and the centrifugal force of the crank and connecting rod. The crank can be readily balanced by making it extend as far on one side of the shaft as it does on the other, but the piston and connecting rod are more difficult to balance. We can greatly reduce the effect of jarring if we balance the crank and make the endwise throw come in line with the foundation, which should be heavy enough to absorb the vibration transmitted. In a horizontal engine this endwise throw not being in line with the foundation will cause vibration in the engine itself.

In machines that can be anchored down to a massive foundation a state of defective balance only results in straining the parts and causing needless wear and friction at the crank-shaft bearings and elsewhere, and in communicating some tremor to the ground. The problem of balancing is of much more consequence in locomotive and marine engines.

To sum up the general advantages of vertical engines: they have less cylinder wear, they take up less floor space, and they can be better balanced. In addition to these there are certain advantages which vertical engines have for certain kinds of work.

The disadvantages of vertical engines are as follows: The

pressure on the crank-pin is greater during the *down* stroke than during the *up* stroke because, during the down stroke the weight of the reciprocating parts is added to the steam pressure, and during the up stroke this weight is subtracted.

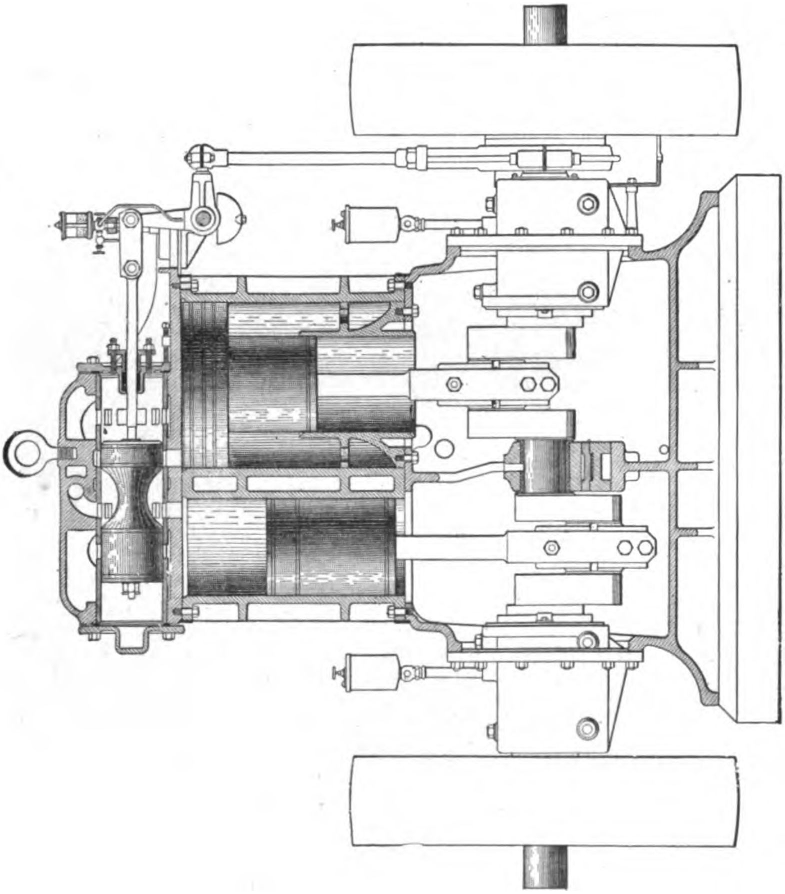


Fig. 18.

Another difficulty is that in large engines the various parts are on such different levels that they require considerable climbing. This requires more attendants and is sometimes the cause for neglect of the engine.

The foundations for vertical engines generally need to be *deeper* than those for horizontal engines. At the same time, however, they need not be as *broad*.

Marine Engines. The first steam vessels were fitted with paddle-wheels, and as beam engines were the most common, this form of engine was used. Its construction, however, was somewhat modified for this service. This arrangement of beam engine and paddle-wheel was used for many years and was applied to ocean vessels as well as to small river boats. It is still used, especially in this country, on river steamers and some coast steamers. The beam is supported by large A frames on the deck, and the engines are about on a level with the shaft.

Engines of this type take up rather more room than those now in common use, partly because of great size, and also because of the shaft and paddle-wheels. Another disadvantage is that in heavy weather, when one paddle-wheel is thrown out of water the other is deeply immersed and takes all the strain, so that there is a tendency to rack the boat. Then again if the boat is loaded heavily the paddle-blades are very deeply immersed while if light they barely touch the water. It is hard to handle the engines satisfactorily under both conditions.

The introduction of the screw propeller overcame these difficulties very largely and at the same time required a much faster running engine. At first, the increased speed was supplied by the use of spur-wheel gearing, but gradually higher speed engines were built and connected directly to the propeller shaft. It was, of course, difficult with small width at each side of the shaft to use horizontal engines, therefore various arrangements of inclined engines were used before the vertical engine was finally chosen by all as the standard form for marine work. It is only in recent years that the vertical engine has become general in naval work and in merchant steamers.

In merchant ocean steamers the common form has three cylinders set in line, fore and aft, above the shaft. The cranks are 120° apart, to give a very even turning moment. The three cylinders are worked *triple expansion.* The valves are usually piston valves on the high and intermediate, and double ported slide valves on the low. Sometimes piston valves are used on all

the cylinders. Plain slide valves are not suitable for high-pressure work of any kind.

For engines on ocean vessels it is necessary to use surface condensers in order that the same water may be used over and over again. If it were necessary to take in sea-water for the boilers they would soon become clogged with the salt and require cleaning. Surface condensers for marine work are generally made up of a large number of brass tubes of from $\frac{5}{8}$ inch to 1 inch in diameter. In some cases the cold water is forced to flow through the tubes while the steam comes in contact with the outside of the tubes.

In any marine plant there are four special pumps. The *first* is the air pump for the condenser. This is usually made large so that in case there is a leak in the condenser it can take charge of the water even if it becomes necessary to run as a jet instead of a surface condenser. The *second* is the feed pump for the boilers. The *third* is the circulating pump, which forces the current of cold water through the condenser. The last is the bilge pump, which pumps out water that gathers in the bilge of the ship by leakage or otherwise. In case of a serious leak all the pumps can be made to pump from the bilges. In some old types all these pumps were worked from the main engine; generally, however, the feed pump and the circulating pump are separate, as also the bilge pump. The circulating pump is, in many modern engine rooms, of the centrifugal type.

Locomotive Engines. Of all steam engines the most inefficient is the steam locomotive. In the first place, the boiler must be forced so hard that the products of combustion pass off at a very high temperature and consequently carry away a great deal of heat. Bits of entirely unburned or partially burned coal are drawn through and wasted.

In the second place, the boiler is exposed to great loss of heat by radiation. Although its surface is lagged, it cannot be very effectively covered, and the wind takes away a great deal of heat.

Mechanically also the locomtive is very imperfect. In most good steam engines the efficiency, that is to say, the ratio of the *effective horse-power* to the *developed horse-power is fully* $\frac{9}{10}$ or 90 per cent. In the locomotive this ratio was shown to be about 43

per cent by two independent tests. This is in part due to the special difficulties in locomotive construction, but the principal losses are those caused by radiation and the escape of heat from the stack.

As to locomotive boilers, Mr. Forney says, "The weight and dimensions of locomotive boilers are in nearly all cases determined by the limits of weight and space to which they are necessarily confined." It may be stated generally that within these limits a locomotive boiler cannot be made too large. In other words, boilers for locomotives should always be made as large as possible under the conditions that determine the weight and dimensions of the locomotives.

There are certain types of locomotives common in American practice which have special names. The eight-wheel or "American" passenger type of locomotive has four coupled driving-wheels and a four-wheeled truck in front. The "ten-wheel" type has six coupled drivers and a leading four-wheel truck. This type is used for both freight and passenger service. The "Mogul" type is used altogether for freight service; it has six coupled drivers and a two-wheel or pony truck in front. The "Consolidation" type is used for heavy freight service. It has eight coupled driving-wheels and a pony truck in front. There are also a great many special types for special purposes. In switch-yards a type of engine is used which has four or six drivers with no truck. The Forney type has four coupled driving-wheels under the engine and a four-wheel truck carrying the water-tank and fuel. This type is used on elevated roads largely. "Decapod" engines are a type used for heavy freight service, having ten coupled driving-wheels and a two-wheel truck in front. A tank engine is one which carries the feed water in tanks on the engine itself instead of in the tender, as in other engines. The various different forms are too numerous even to name.

There has been some effort made to introduce compounding in locomotive practice. It has in some cases been very successful, especially for express trains. A committee of the American Railway Master Mechanics Association says of compounding:

"(*a*) It has achieved a saving in the fuel burnt, averaging 18 per cent at reasonable boiler pressures.

(*b*) It has lessened the amount of water to be handle.

(*c*) The tender can, therefore, be reduced in size and weight.

(*d*) It has increased the possibilities of speed beyond sixty miles per hour, without unduly straining the engine.

(*e*) It has increased the haulage power at full speed.

(*f*) In some classes of engines it has increased the starting power.

(*g*) It has lessened the valve friction per horse-power developed."

A number of other reasons are given in their report.

In opposition to this may be mentioned the extra first cost of the engine and the cost of maintaining a more complicated machine. It is much more work to keep it in repair and many engineers are of the opinion that the saving in fuel is only sufficient to offset these extra expenses. If the engine is running under steady load, the compounding will effect a great saving; but in many parts of the country the load varies constantly, due to grades in the road.

We shall learn later that a compound engine cannot work efficiently under light load. If the grades are first up and then down, the simple engine is the more economical. For a steady up-grade the compound is more economical, as it can be run steadily under full load. This is especially true in mountain districts, where the long up-grades and scarcity of fuel and water make ideal conditions for compound locomotives. Through freight service probably offers the widest field for success with these engines.

It has already been said that it is difficult to balance an engine completely. This defect is very much greater in locomotives than in stationary engines. Lack of balance in a locomotive results in serious pounding of the track. Also there is danger of flattening and breaking the wheels, and the rails may be seriously damaged.

Pumping Engines. The first steam engines built were pumping engines and today the most economical engines are those built for this work.

In pumping engines it is not absolutely necessary to have a revolving shaft. All that is required is that the piston in the pump cylinder shall be driven back and forth with a plain recipro-

cating motion which may be exactly like that of the steam piston. For this reason, in early pumping engines and also in many modern engines, the reciprocating motion of the steam piston is applied directly, or through a beam, to produce the reciprocating motion of the pump piston or plunger without the use of any revolving part. Frequently, however, it is desirable to use a fly wheel so that the steam may be used expansively, and in these cases, of course, a revolving shaft must be used. Fig. 19 shows a power pump.

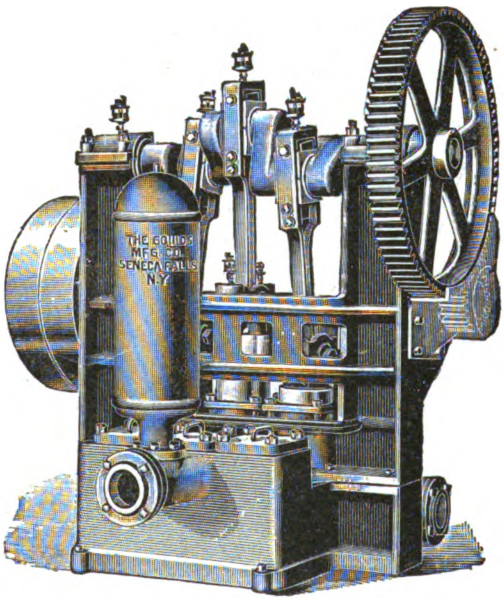


Fig 19.

For deep-well or mine pumping, the cylinders are often set in a vertical position directly over the pump cylinder. The piston rod extends from the steam cylinder directly below to the pump plunger. Sometimes it is possible to use steam expansively in these pumps by reason of the weight of the reciprocating parts. When the weight is sufficient, the steam can be cut off before the end of the stroke and the momentum of the parts will be enough to just finish the stroke, consequently these pumps are sometimes compounded. They are possible only in pumping from very deep wells.

Direct-acting Steam Pumps. Fly wheel pumps have one disadvantage, if run too slowly the momentum of the fly wheel is not sufficient to carry it by the dead centers; if run too fast the flv wheel is in danger of bursting. A fly-wheel pump can be made to discharge a small amount of water by means of a by-pass valve, but of course it then runs at a disadvantage.

The direct-acting steam pump shown in Fig. 20 has the steam piston at one end of a rod and the water piston at the other end. The steam pressure acts directly on the piston; no fly wheel is used, and since the reciprocating parts are comparatively light, and there is no revolving mass to carry by the dead points, it is evident that in the ordinary form there can be no expansion of steam. The pump is inexpensive and gives a positive action. It uses a great quantity of steam relatively, but for small work the absolute amount is not very great. Even in larger engines the lighter foundations that are possible and the slight first cost are frequently controlling features.

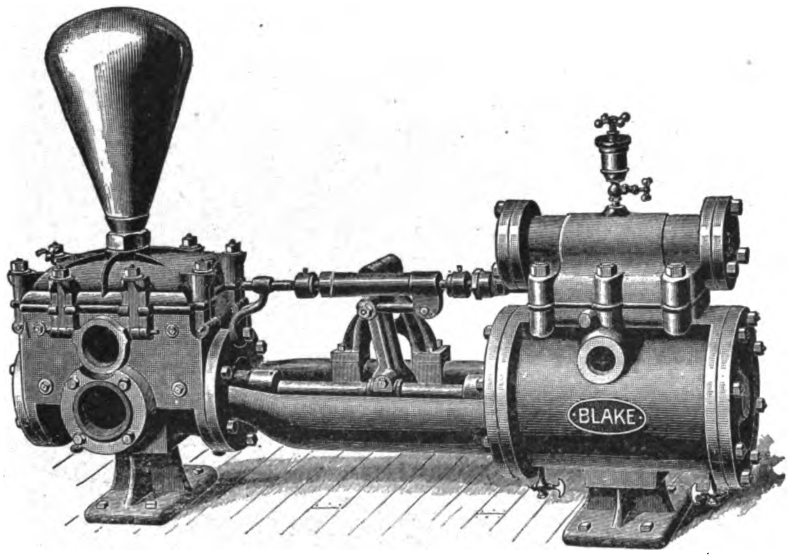


Fig. 20.

A rocker or bell-crank lever on the piston rod moves the steam valve and admits steam to the other side of the piston while opening the first side to exhaust. In large pumps of this kind, and even in some small ones, this motion merely admits steam to a small auxiliary piston which then moves the main steam valve by steam pressure. Some pumps operate the steam valve by means of a tappet instead of a rocker and bell-crank lever.

There have been various devices tried for using steam expansively in these direct-acting pumps without the use of a fly-wheel.

In order to do this it is necessary to provide some means of storing energy during the early part of the stroke and returning it during the latter part, when, owing to the expansion, the pressure of the steam is less. One such device is as follows: a crosshead A, (Fig. 21) fixed to the piston rod is connected to the plungers of a pair of oscillating cylinders B B, which contain water and communicate with a reservoir full of air compressed to about 300 pounds per square inch. When the stroke (which takes place in the direction of the arrow) begins, these plungers are first forced in, and hence work is at first done on the main piston rod, through the compensating cylinders B B, on the compressed air in the reservoir. This continues until the crosshead has advanced so that the oscillating cylinders stand at right angles to the line of stroke. Then for the remainder of the stroke their plungers

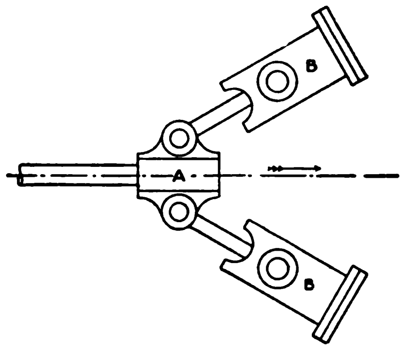


Fig. 21.

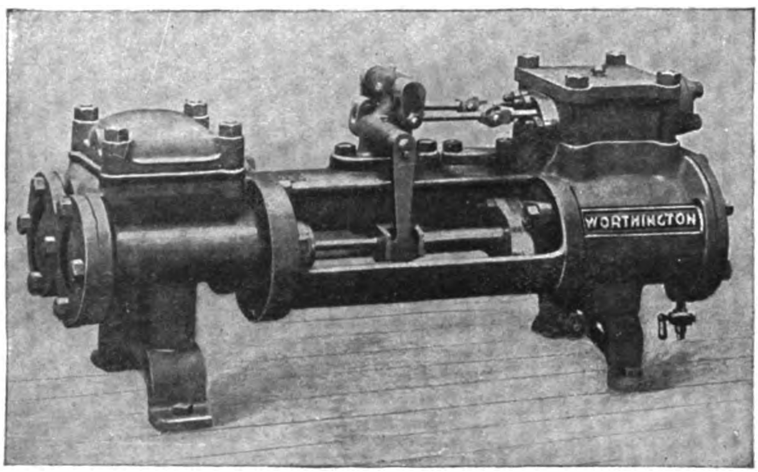


Fig. 22.

assist in driving the main piston, and the compressed air gives out the energy which it stored in the earlier portion.

The Duplex steam pump consists simply of two direct-acting steam pumps placed side by side, as shown in Fig. 22. On the piston rod on one side is a bell-crank lever which operates the valve of the other pump. On the further piston rod is a rocker arm which moves the valves of the first pump. There must be a rocker on one side and a bell-crank lever on the other because of the relative motion of the valves and pistons. The first piston, as it goes forward, must use a rocker because it draws the second valve back. The second piston, as it goes back, must use a bell-crank lever because it must push the first valve back in the same direction as its own motion. The two pistons are made to work a half-stroke apart. Thus one begins its stroke when the other is in the

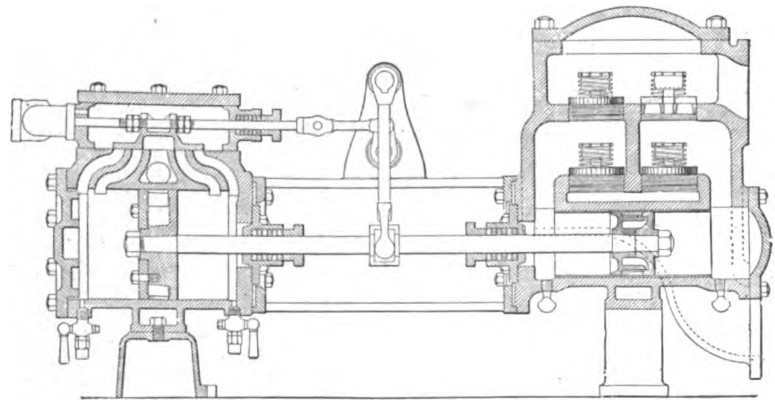
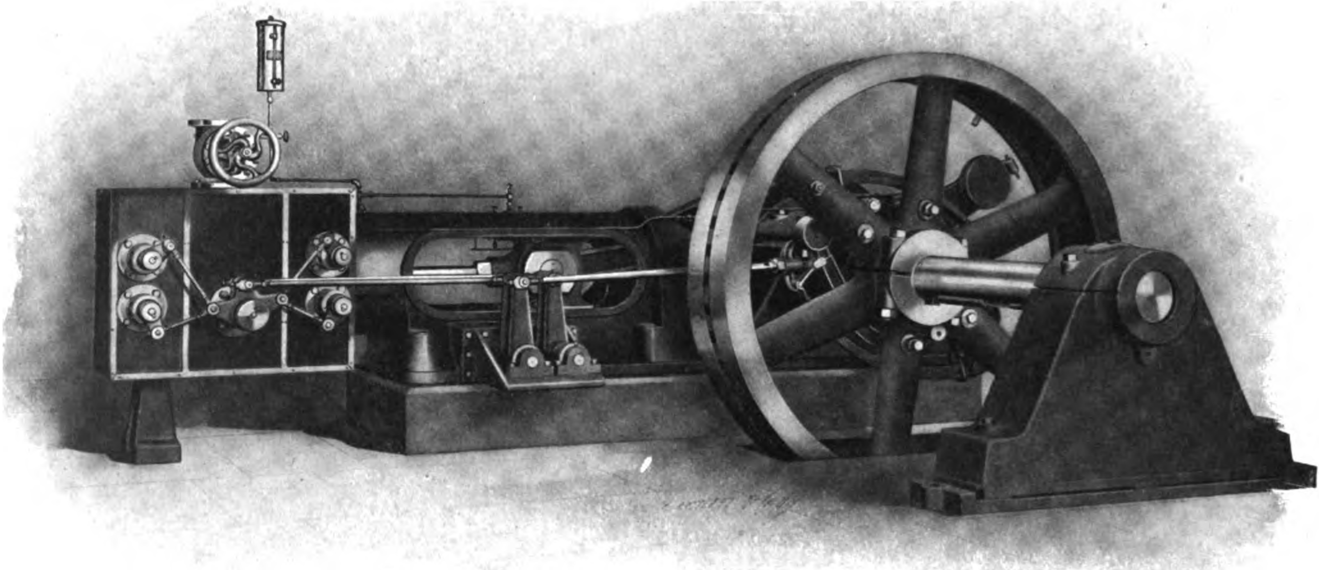


Fig. 23.

middle. In this way a steadier flow of water is obtained, for both pumps discharge into the same delivery pipe. The pumps may be made compound. A sectional view of the pump is shown in Fig. 23.

Corliss Engines. In large engines a common way of regulating the steam supply is by changing automatically the point in the stroke of the piston at which the steam is cut off. This is frequently accomplished by using some trip gear similar to the one first introduced by Geo. Corliss. These gears are called Corliss gears. In the Corliss gear there is a separate admission valve and a separate exhaust valve for each end of the cylinder, as shown in Fig. 24. The exhaust valves are opened and closed by



SIMPLE CORLISS VALVE ENGINE.
Ball and Wood Co.

the motion of rods or cranks connected to them. The admission valves are opened in the same way, but they snap shut by themselves when the piston has reached a certain point of its stroke. This point will vary with the position of the governor, which in turn depends on the speed of the engine. These Corliss engines cannot be run at high speed because the trip gear requires some time to act.

The valves of Corliss engines turn in hollow cylindrical seats which extend across the cylinders. A wrist plate which turns on

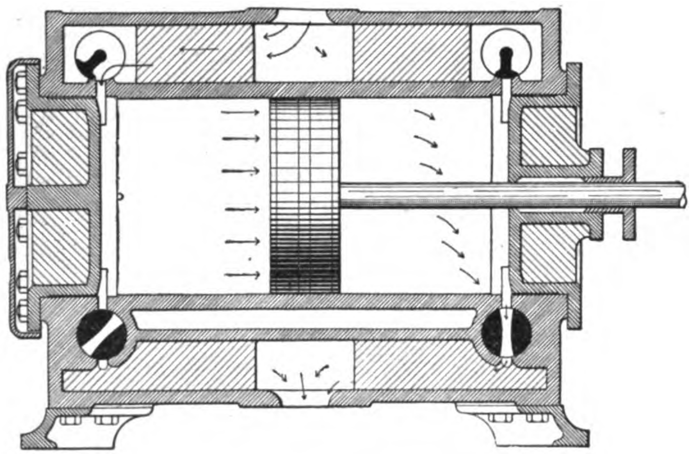


Fig. 24.

a pin on the outside of the cylinder receives a motion of oscillation from an eccentric and opens the valves by means of the rod connections. When the piston has reached a point where the steam should be cut off, the trip gear is held in such a position by the governor that it releases the valve, which now springs shut under the action of the dash-pot. The admission valve to the other side of the cylinder is controlled in exactly the same way.

The admission valves are generally placed at the top and the exhaust valves at the bottom of the cylinder. Any water which may be formed by steam condensing can readily drain off by this arrangement. There are a great many modifications of the Corliss gear. Fig. 25 is a Harris Corliss Engine.

The advantage of the Corliss gear is the great range through

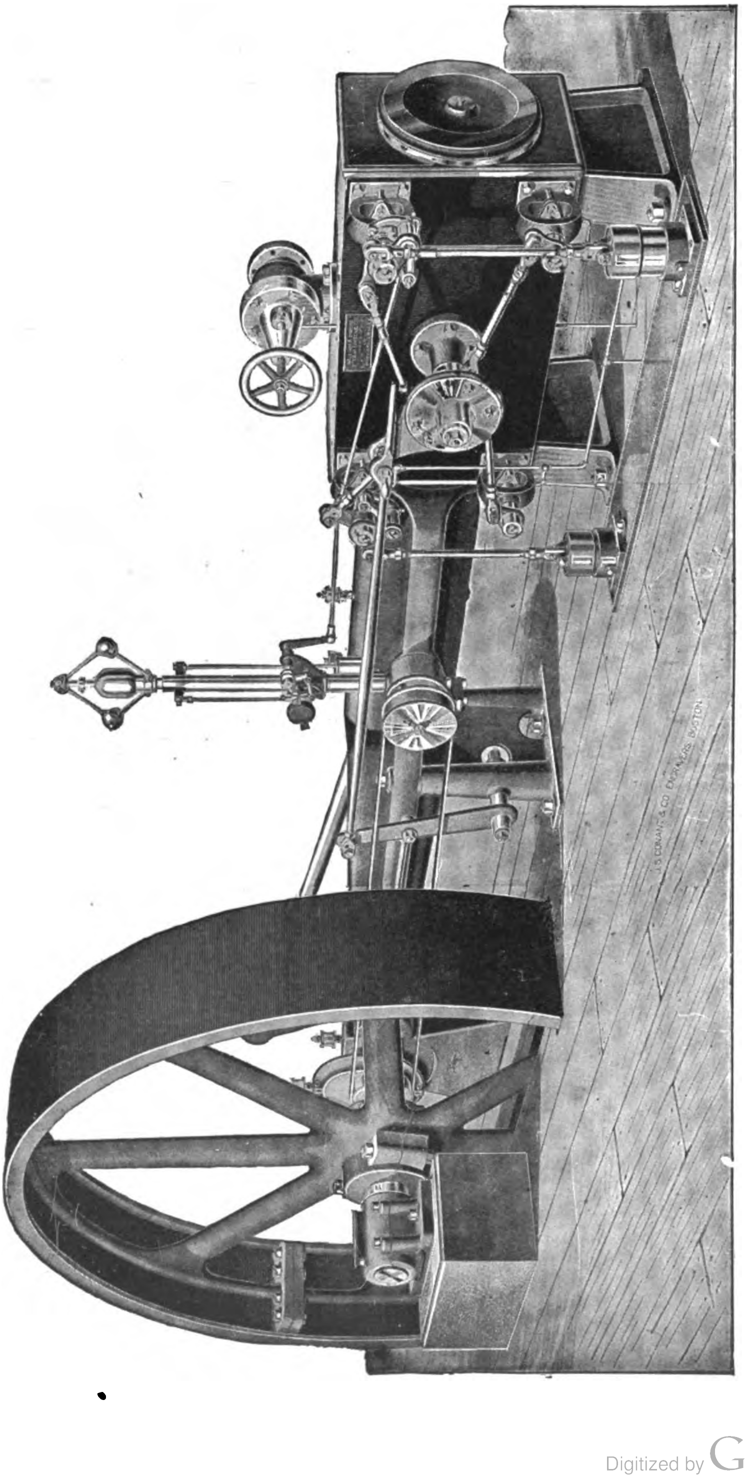


Fig. 25.

which the cut-off can be varied, from very early to very late in the stroke. Another great advantage is the quick action which reduces wire drawing. To understand fully the loss caused by wire drawing requires a knowledge of higher mathematics.

CONDENSERS.

When low-pressure steam is cooled it gives up its latent heat; that is, it changes from a vapor to a liquid. We know that a liquid occupies much less space than an equal weight of vapor; hence, by changing the steam to water the pressure is greatly reduced. By cooling the steam in the cylinder in front of the piston the back pressure, or resistance, is decreased. This reduces the pressure necessary to push the piston through the stroke, therefore less steam is required to do the work. This cooling is accomplished by some form of condenser.

There are two types of condensers, surface and jet. A surface condenser is one in which the steam passes through pipes surrounded by water or the water flows through pipes surrounded by steam. In the jet condenser the steam is condensed by coming in contact with cold water, which enters as spray. In both types the steam is condensed to water and a partial vacuum is formed, because water occupies much less space than does an equal weight of steam. If it were not for the air in the entering steam there would be an almost perfect vacuum. For this reason every condenser is fitted with an air pump to remove this air and the condensed steam.

Surface Condensers. The condenser shown in section in Fig. 26 is a common form of the surface type, in which the air pump and circulating pump are both direct acting and are operated by the same steam cylinder. The cold condensing water is drawn from the supply into the circulating or water pump. This pump forces the water up through the valves and water inlet to the condenser. It flows, as indicated by the arrows, through the inner tubes of the lower section, then back through the space between the inner and outer tubes. The water then passes upward and through the upper section, as it did in the lower. It then passes out of the condenser through the water outlet, taking with it the heat it has received from the steam.

The exhaust steam from the engine enters at the exhaust inlet and comes in contact with the perforated plate, which scatters it among the tubes. This method protects the upper tubing from the effect of direct contact with the exhaust steam. The steam expanding in the condenser comes in contact with the cool tubes,

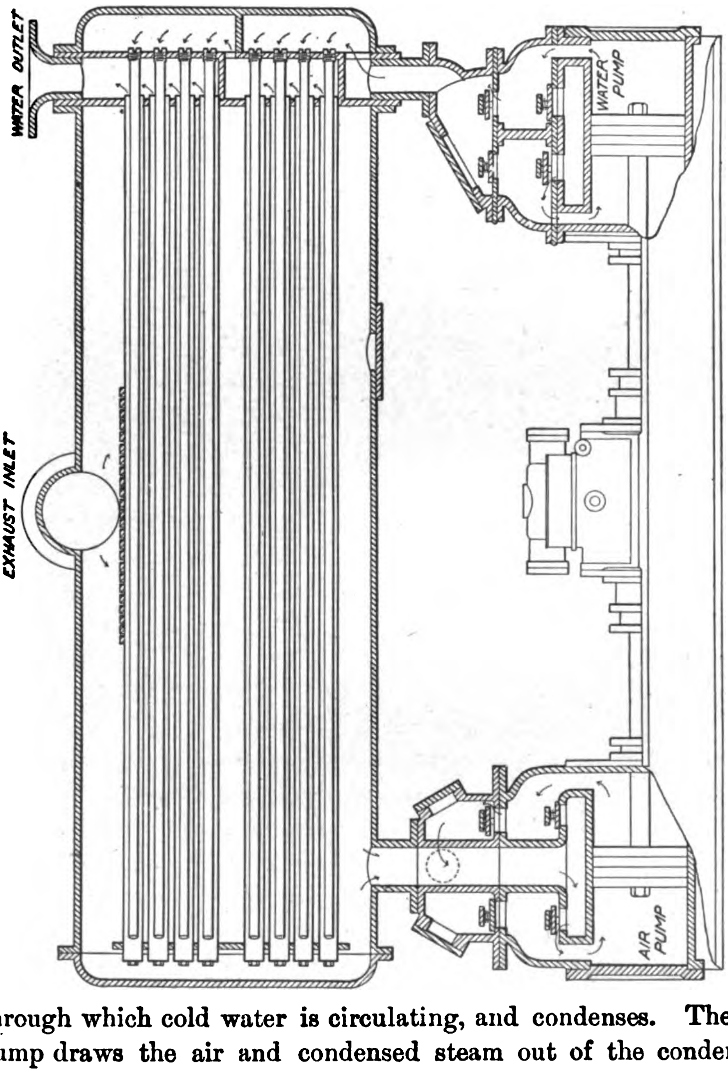


Fig. 26.

through which cold water is circulating, and condenses. The air pump draws the air and condensed steam out of the condenser and thus maintains a partial vacuum. This causes the exhaust steam in the engine cylinder to be drawn into the condenser.

The condensed exhaust steam collects at the bottom of the condenser, is drawn into the air pump cylinder and is discharged while heated to the hot well of the boiler. The use of this hot water as feed water is a considerable saving; but the great advantage of the condenser is the reduction in back pressure.

Hot water cannot be used by an ordinary pump as well as cold water because of the pressure of the vapor which arises from the hot water. In the condenser shown, the water and air pumps are run by the piston in the steam cylinder. Sometimes these pumps are connected to the main engine and receive motion from the shaft or crosshead.

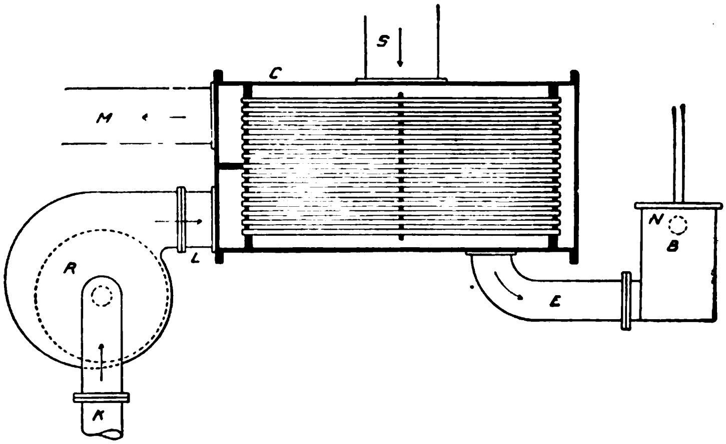


Fig. 27.

The general arrangement of the surface condenser with the necessary pumps is shown in Fig. 27. The cooling water enters through the pipe K, and flows to the circulating pump R, which forces the water into the condenser through the pipe L. In case the water enters the condenser under pressure from city mains no circulating pump is necessary. After flowing through the tubes it leaves the condenser by means of the exit M, and flows away. Exhaust steam enters at S, and is condensed by coming in contact with the cold tubes; the water (condensed steam) then falls to the bottom of the condenser and flows to the air pump B by the pipe E. The air pump removes the air, vapor and condensed steam from the condenser and forces it through the pipe N into the hot well, from which it goes to the boilers or to the feed tank.

The circulating pump, when separate from the condenser, is usually of the centrifugal type. This pump consists of a fan or wheel which is made up of a central web or hub, and arms or vanes. This pump is shown in Fig. 28. The vanes are curved, and as the water is drawn in at the central part the vanes throw it off at the circumference. A suitable casing directs the flow. This type of pump is advantageous because there are no valves to get out of order, and as the lift is little, if any, the pump will discharge a large volume of water in a nearly constant stream. The circulating pump is usually so placed that the water flows to it under a slight head. The pump is driven by an independent engine so that the circulating water may cool the condenser even if the main engine is not working.

The centrifugal pump works more smoothly and with less trouble than an ordinary force pump, because it is not reciprocating and it has no valves.

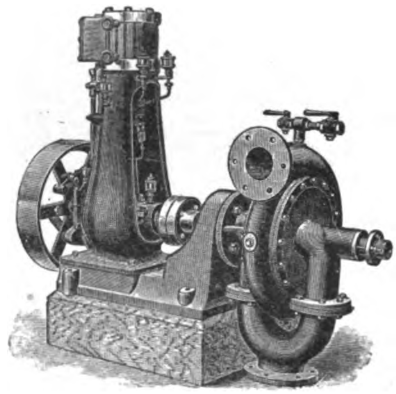


Fig. 28.

Jet Condensers. Fig. 29 shows the longitudinal section of an independent jet condenser with the pump. The cold water used to condense the steam enters at A, passes down the spray pipe B, and is broken into a fine spray by means of the spray cone C. This action insures a rapid and thorough mixing of the steam and water and consequently a rapid condensation. The exhaust steam enters at D, with a comparatively high velocity, which is imparted to the water. The whole mixture of water, steam and vapor passes at high velocity through the conical chamber E to the pump cylinder F. The pump forces this water to the pipe G. The spray cone is adjusted by the stem which passes through the stuffing box at the top of the condenser. The valves are shown at H and K. The steam end of the pump is at L. Motion of the valve is produced by the rocker arm J.

In Fig. 30 a jet condenser is shown connected to a stationary engine and boiler. The exhaust pipe leads from the engine to the

condenser, the arrows indicating the direction of flow. Cold water enters the condenser through a pipe connected to the well. Part of the mixture of exhaust steam and condensed water goes to the feed-water heater, which is kept nearly full; the rest passes to the sewer. The heater is placed a little above the feed pump, so that the water will enter the pump under a slight head, because the pump cannot raise water warmed by exhaust steam as readily as cold water.

The surface condenser is used almost universally in marine practice. Its first cost is more than that of the jet condenser and it requires more condensing water, but it allows only the condensed steam to return to the boiler. It is also used in stationary work when the condensing water is very impure. The jet condenser is not adapted for marine work, as it pumps both the condensing water and the condensed steam to the hot well. Hence, if salt water or water containing lime is used, it will enter the boiler and form sediment and scale. This type is used

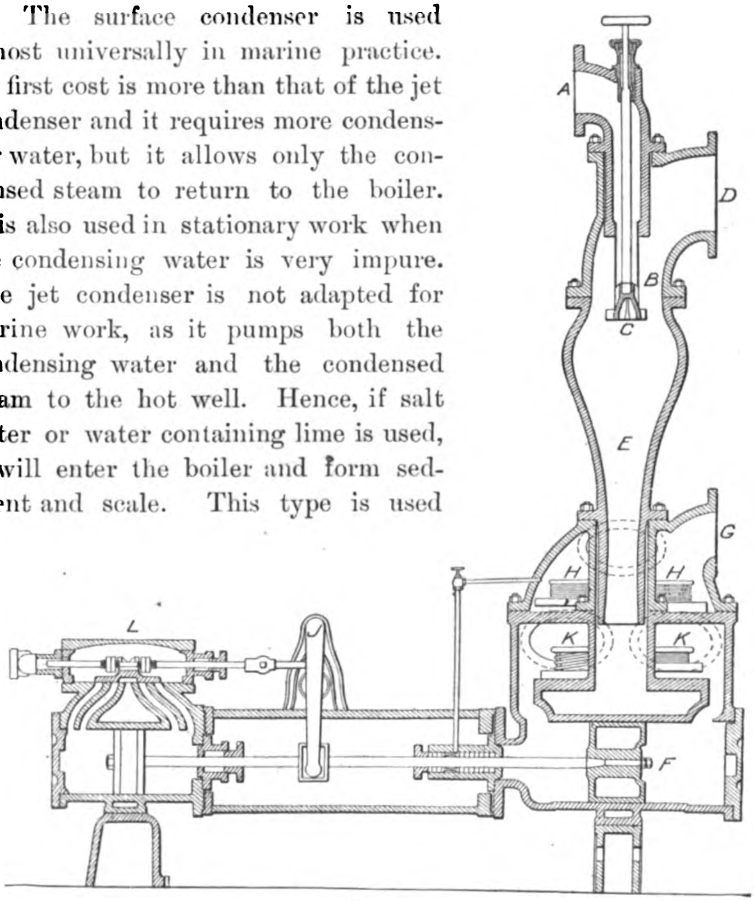


Fig. 29.

where fresh and moderately pure water is available.

It has been mentioned before that Watt always condensed the exhaust steam from his engines, and that when higher pres-

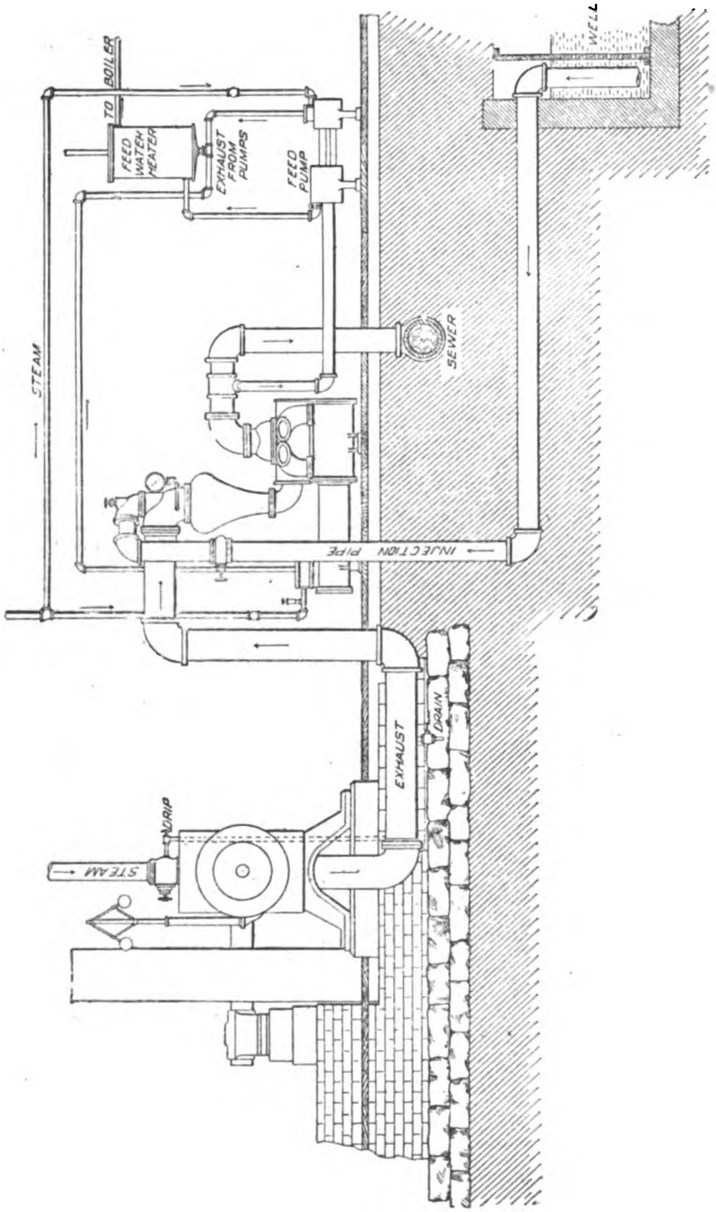

TO BOILER
FEED WATER HEATER
EXHAUST FROM PUMPS
FEED PUMP
STEAM
SEWER
INJECTION PIPE
WELL
DRIP
STEAM
EXHAUST
DRAIN

Fig. 30.

sures came into use some makers let the steam discharge into the atmosphere. This leads to the distinction between condensing and noncondensing engines. Both types are in common use, but the condensing engines are much more economical than the noncondensing, as far as fuel is concerned; but to condense the steam, considerable water is necessary, and condensing engines cost more and require more care. Consequently in some cases it is quite as economical, all things considered, to use the noncondensing engine.

COOLING TOWERS.

It sometimes happens that it is impossible to place a steam plant in close proximity to a natural water supply. In such cases the water necessary for the condenser (the circulating water) is expensive, and if the cost is very great it does not pay to add the condenser, because the cost of the circulating water might more than offset the gain from condensing. If, however, some means could be provided whereby the circulating water as it issues from the condenser could be cooled and then used over again in the condenser, the noncondensing engine could then be run condensing; thus taking advantage of all the benefits due to the use of reduced back pressure and heating of the feed water. This has been attempted by conducting the heated discharge water to a pond, where it is allowed to cool to a lower temperature before being used again. Another plan is to place in the yard or on the roof of the building large shallow pans, in which the water is cooled by being exposed to the atmosphere. These methods are unsatisfactory on account of the considerable area necessary and the slow action. In addition they are uncertain, because they are dependent upon atmospheric conditions.

A more efficient and at the same time more expensive process is to use a cooling tower or a water table. Fig. 31 shows the general arrangement of a cooling tower located upon the roof of a building. The discharge from the condenser is led, as shown by the arrows, to the top of the cooling tower, where it is cooled before being returned to the condenser. This cooling is effected by distributing the water, by a system of piping, to the upper edge of a series of mats or slats, over the surface of which the water

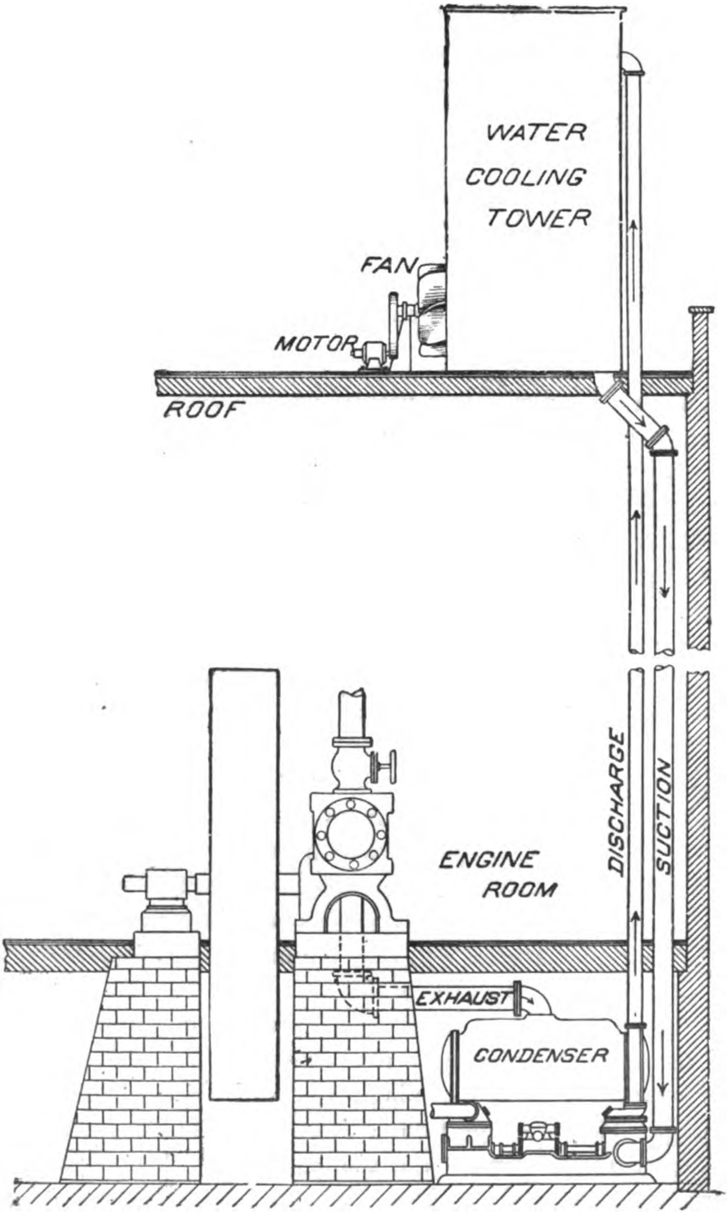

WATER
COOLING
TOWER
FAN
MOTOR
ROOF
DISCHARGE
SUCTION
ENGINE
ROOM
EXHAUST
CONDENSER

Fig. 81.

flows in a thin film to a reservoir which is situated in the bottom of the cooling tower. The mats partially interrupt the flow, and by breaking up the water in small streams cause new portions to be exposed to the cooling effect of the air currents. The water from the reservoir then flows downward through the suction pipe, and is pumped by the circulating pump through the condenser. After passing through the condenser and absorbing heat from the exhaust steam, it rises through the discharge pipe and commences the circuit over again.

The tower may have several arrangements and be made of various materials. A satisfactory form is constructed of steel plates; within the tower are a large number of mats of steel wire cloth galvanized after weaving.

To assist in the cooling of the water, the air is often made to circulate rapidly by means of a fan, which forces the air into the lower part of the tower and upwards among the mats. This fan is usually of the ordinary type, and may be driven by an electric motor, a line of shafting, or by a small independent engine.

In case the fan is not used, the mats are arranged so that they are exposed to the atmosphere, as shown in Fig. 32. This of course necessitates the removal of the steel casing. Usually the fanless tower must be placed at the top of a high building, or in some position where the currents of air can readily circulate among the mats.

With an efficient type of cooling tower the water may be reduced from 30 to 50°, thus allowing a vacuum of from 22 to 26 inches. This will of course greatly increase the economy of the plant, and allow the heated feed water to be returned to the boiler

The water table is usually made of wooden slats placed in the ground near the plant. After trickling over the slats and becoming cooled by the air, it collects in the bottom of the reservoir and is then pumped into the condenser.

THE FLY WHEEL.

It is evident that while the piston can push the crank around during part of the stroke, and pull it during another part, there are still two places (called dead points) at the ends of the stroke, where the pressure on the piston, no matter how great, can exert

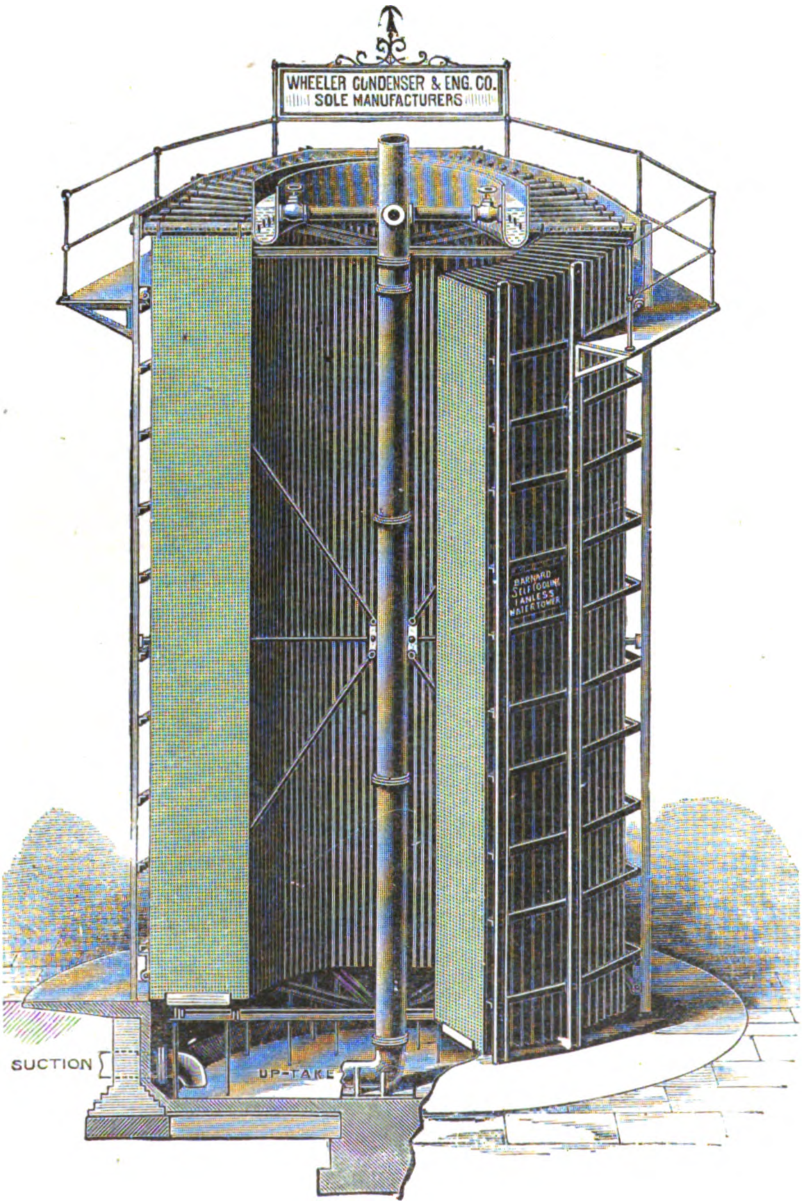

WHEELER CONDENSER & ENG. CO.
SOLE MANUFACTURERS
SUCTION
UP-TAKE

Fig. 32

no turning moment on the shaft. Therefore, if some means is not provided for making the shaft turn past these points without the assistance of the piston, it may stop. This means is provided in the fly wheel, which is merely a heavy wheel placed on the shaft. On account of the momentum of the fly wheel it cannot be stopped quickly, and therefore carries the shaft around until the piston can again either push or pull.

If a long period be considered, the mean effort and the mean resistance must be equal; but during this period there are temporary changes of effort, the excesses causing increase of speed. To moderate these fluctuations several methods are employed.

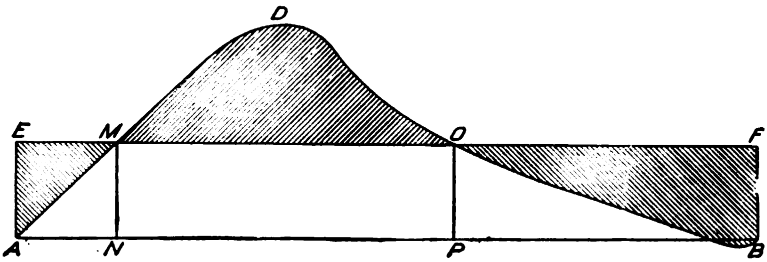


Fig. 33.

The turning moment on the shaft of a single cylinder engine varies, first, because of the change in steam pressure, and second, on account of the angularity of the connecting rod. Before the piston reaches mid-stroke the turning moment is a maximum, as shown by the curve, (Fig. 33). While near the ends of the stroke (the dead points) the turning moment diminishes and finally becomes zero. This, of course, tends to cause a corresponding change in the speed of rotation of the shaft. In order to have this speed as nearly constant as possible, and to give a greater uniformity of driving power, the engine may be run at high speed. By this means the inertia of the revolving parts, such as the connecting rod and crank, causes less variation. When the work to be done is steady and always in the same direction (as in most factories), a heavy fly wheel may be used.

The heavier the fly wheel, the steadier will be the motion. It is, of course, desirable in all factory engines to have steady motion, but in some it is more important than in others. For

instance, in a cotton mill it is absolutely necessary that the machinery shall move with almost perfect steadiness; consequently mill engines always have very large, heavy fly wheels. It is undesirable to use larger wheels than are absolutely necessary, because of the cost of the metal, the weight on the bearings and the danger from bursting.

If the turning moment which is exerted on the shaft from the piston could be made more regular, and if dead points could be avoided, it would be possible to get a steadier motion with a much smaller fly wheel.

If the engine must be stopped and reversed frequently, it is better to use two or more cylinders connected to the same shaft. The cranks are placed at such angles that when one is exerting its minimum rotative effort, the other is exerting its maximum turning moment; or, when one is at a dead center, the other is exerting its greatest power. These two cylinders may be identically the same, as is the case with most hoisting engines and with many locomotives; or the engine may be compound or triple expansion.

This arrangement is also used on engines, for mines, collieries, and for hoisting of any sort where ease of stopping, starting and reversing are necessary. Simple expansion engines with their cranks at right angles are said to be coupled.

The governor adjusts the power of the engine to any large variation of the resistance. The fly wheel has a duty to perform which is similar to that of the governor. It is designed to adjust the effort of the engine to sudden changes of the load which may occur during a single stroke. It also equalizes the variation in rotative effort on the crank pin. The fly wheel absorbs energy while the turning moment is in excess of the resistance and restores it while the crank is at or near the dead points. During these periods the resistance is in excess of the power.

The action of the fly wheel may be represented as in Figs. 33 and 34. It will be noticed that in Fig. 33 the curve of crank effort runs below the axis toward the end of the stroke. This is because the compression is greater than the pressure near the end of expansion, and produces a resultant pressure on the piston. In Fig. 34 the effect of compression has been neglected. Let us suppose that the resistance, or load. is uniform. In Fig. 33 the line

A B is the length of the semi-circumference of the crank pin, or it is the distance the crank pin moves during one stroke. The curve A M D O B is the curve of turning moment for one stroke. M N is the mean ordinate, and therefore A E F B represents the constant resistance. The effort and resistance must be equal if the speed is uniform; hence A E F B = A M D O B. Then area A E M + area O F B = area M D O. At A the rotative effort is zero because the crank pin is at the dead point; from A to N the turning moment is less than the resistance. At N the resistance and the effort are equal. From N to P the effort is in excess of the resistance. At P the effort and resistance are again equal. From P to B the resistance is greater than the effort. In other words, from A to N the work done by the steam is less than the resistance.

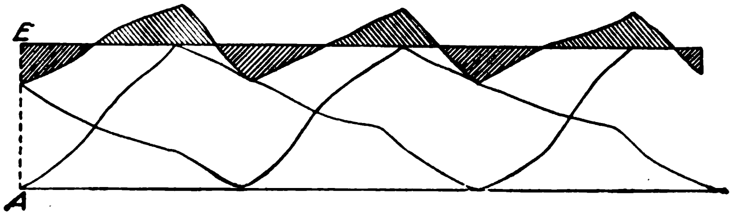


Fig. 34.

This shows that the work represented by the area A E M must have been done by the moving parts of the engine. From N to P the work done by the steam is greater than the resistance, and the excess of energy is absorbed by or stored in the moving parts. From P to the end of the stroke, the work represented by the area O F B is done on the crank pin by the moving parts.

We know from the formula, $E = \frac{W V^2}{2g}$, that energy is proportional to the square of the velocity. Hence as W and g remain the same, the velocity must be reduced when the moving parts are giving out energy, and increased when receiving energy. Thus we see that the tendency of the crank pin is to move slowly, then more rapidly. The revolving parts of an engine have not sufficient weight to store this surplus energy, hence a heavy fly wheel is used.

In case there are two engines at right angles, two effort curves

must be drawn, as shown in Fig. 34. The mean ordinate A E is equal to the mean or constant resistance. There are two minimum and two maximum velocities in one stroke. The diagram shows that the variation is much less than for a single cylinder; hence a lighter fly wheel may be used.

The *weight of the fly wheel* depends upon the character of the work done by the engine. For pumping engines and ordinary machine work the effort need not be as constant as for electric lighting and fine work. In determining the weight of a fly wheel the diameter of the wheel must be known, or the ratio of the diameter of the wheel to the length of stroke. If the wheel is too large, the high linear velocity of the rim will cause too great a centrifugal force and the wheel is likely to break. In practice, about 6,000 feet per minute is taken as the maximum linear velocity of cast-iron wheels. When made of wood and carefully put together, the velocity may be taken as 7,000 to 7,500 feet per minute.

We know that linear velocity is expressed in feet per minute by the formula, $V = 2\pi R N$, or $V = \pi D N$.

Then if a wheel runs at 100 revolutions per minute, the allowable diameter would be,

$$6{,}000 = 3.1416 \times D \times 100$$

$$D = \frac{6{,}000}{3.1416 \times 100} = 19.1 \text{ feet.}$$

If a wheel is 12 feet in diameter the allowable speed is found to be,

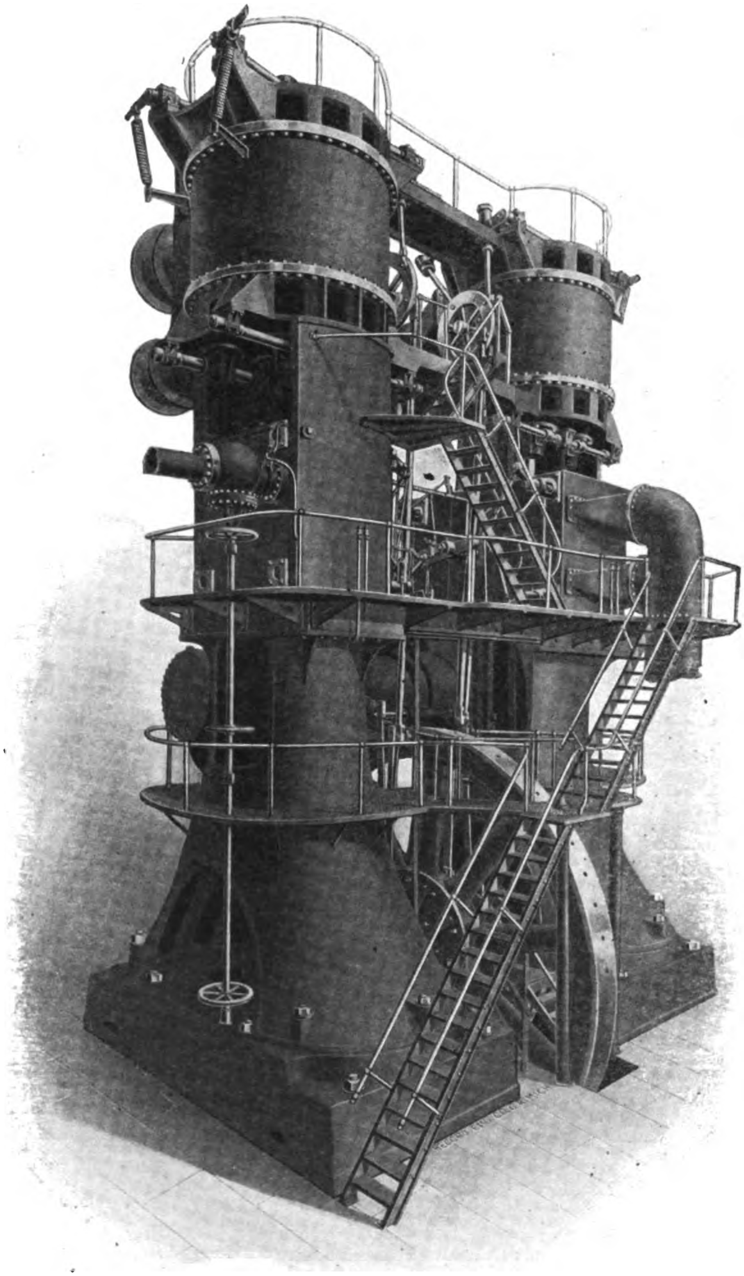
$$N = \frac{V}{\pi D}$$

$$= \frac{6{,}000}{3.1416 \times 12} = 159 \text{ revolutions per minute.}$$

It is usual to make the diameter a little less than the calculated diameter.

Having determined the diameter, the weight may be calculated by several methods. There are many formulas to obtain this result given by various authorities. One formula is given as follows:

$$W = \frac{C \times d^2 \times b}{D^2 \times N^2}$$



BLOWING ENGINE.

Steam Cylinder 58 × 108 × 60 inches. Air Cylinders 108 inches diameter.

Total Weight of Engine 1,270,000 Pounds.

William Tod Company.

In the above, $W =$ weight of the rim in pounds
$d =$ diameter of the cylinder in inches
$b =$ length of stroke in inches
$D =$ diameter of fly wheel in feet
$N =$ number of revolutions per minute

C is a constant which varies for different types and conditions.

Slide-valve engines, ordinary work,	$C = 350,000$
Corliss engines, ordinary work,	$C = 700,000$
Slide-valve engines, electric lighting,	$C = 700,000$
Corliss engines, electric lighting,	$C = 1,000,000$
Automatic high-speed engines,	$C = 1,000,000$

Example. Let us find the weight of a fly-wheel rim for an automatic high-speed engine used for electric lighting. The cylinder is 24 inches in diameter; the stroke is 2 feet. It runs at 300 revolutions per minute, and the fly wheel is to be 6 feet in diameter.

$$W = 1,000,000 \times \frac{(24)^2 \times 24}{36 \times 90,000}$$

$$W = 4,266 \text{ pounds}$$

Another example. A plain slide-valve engine for electric lighting is 20″ × 24″. It runs at 150 revolutions per minute. The fly wheel is to be 8 feet in diameter. What is the weight of its rim?

$$W = 700,000 \times \frac{400 \times 24}{64 \times 22,500}$$

$$W = 4,666 \text{ pounds.}$$

The weight of a fly wheel is considered as being in the rim. The weight of the hub and arms is simply extra weight. Then, if we know the weight of the rim and its diameter, we can find the width of face and thickness of rim. Let us assume the given diameter to be the mean of the diameter of the inside and outside of the rim.

Let $b =$ width of face in inches
$t =$ thickness of rim in inches
$d =$ diameter of fly wheel in inches
$.2607 =$ weight of 1 cubic inch of cast iron

Then,

$$W = .2607 \times b \times t \times \pi d$$
$$= b \times t \times .819\, d$$

Suppose the rim of a fly wheel weighs 6,000 pounds, is 9 feet in diameter, and the width of the face is 24 inches. What is the thickness of the rim?

$$t = \frac{W}{.819\,d\,b}$$

$$= \frac{6{,}000}{.819 \times 108 \times 24}$$

$$= 2.83 \text{ inches}$$

In this case the rim would probably be made $2\frac{13}{16}$ inches thick. The total weight, including hub and arms, would probably be about 8,000 pounds.

GOVERNORS.

The load on an engine is never constant, although there are cases where it is nearly uniform. While the engine is running at constant speed, the resistance at the fly-wheel rim is equal to the work done by the steam. If the load on the engine is wholly or partially removed, and the supply of steam continues undiminished, the force exerted by the steam will be in excess of the resistance. Work is equal to force multiplied by distance; hence, with constant effort, if the resistance is diminished, the distance must be increased. In other words, the speed of the engine will be increased. The engine will "race," as it is called. Also, if the load increases and the steam supply remains constant, the engine will "slow down."

It is evident, then, that if the speed is to be kept constant some means must be provided so that the steam supply shall at all times be exactly proportional to the load. This is accomplished by means of a governor.

Steam-engine governors act in one of two ways: they may regulate the *pressure* of steam admitted to the steam chest, or they may adjust the speed by altering the *amount* of steam admitted. Those which act in the first way are called throttling governors, because they throttle the steam in the main steam pipe. Those of the latter class are called automatic cut-off governors, since they automatically regulate the *cut-off*.

Theoretically, the method of governing by throttling the

steam causes a loss in efficiency, but the throttling superheats the steam, thus reducing cylinder condensation. By the second method the loss in efficiency is very slight, unless the ratio of expansion is already great, in which case shortening the cut-off causes more cylinder condensation. This subject will be taken up in detail later.

In most governors, centrifugal force, counteracted by some other force, is employed. A pair of heavy masses (usually iron balls or weights) are made to revolve about a spindle, which is driven by the engine. When the speed increases, centrifugal force increases, and the balls tend to fly outward; that is, they revolve in a larger circle. The controlling force, which is usually gravity or springs, is no longer able to keep the balls in their former path. When, therefore, the increase is sufficiently great, the balls in moving outward act on the regulator, which may throttle the steam or cause cut-off to occur earlier.

With the throttling governor a balanced throttle valve is placed in the main steam pipe leading to the valve chest. If the engine runs faster than the desired speed, the balls are forced to revolve at a higher speed. The increase in centrifugal force will cause them to revolve in a larger circle and in a *higher plane*. By means of some mechanism (levers and gears) the spindle may be forced downward, thus partially closing the valve. The engine, therefore, takes steam at a lower pressure, and the power supplied being less, the speed falls slightly.

Similarly, if the load is increased, the engine slows down, which causes the balls to drop and open the valve more widely, steam at higher pressure is then admitted, and the speed is increased to the regular number of revolutions.

With the Corliss or Wheelock engine the governor of this type acts differently. Instead of throttling the steam in the steam pipe, the governor is connected to the releasing gear by rods. An increase of speed causes the releasing gear to unhook the disengaging link earlier in the stroke. This causes earlier cut-off, which of course decreases the power and the speed, since the amount of steam admitted is less. If for any reason the load increases, the governor causes the valves to be held open longer. The cut-off, therefore, occurs later in the stroke.

One of the most common forms of governor is similar to that invented by James Watt. It is called, from its appearance, the pendulum governor. It is shown in Fig. 35. To consider the theory of the pendulum governor, the masses of the balls are assumed to be concentrated at their centers, and the rods made of some material having no weight.

When the governor is revolving about its axis at a constant speed the balls revolve in a circle having a radius r The distance from this plane to the intersection of the rods, or the rods produced, is called the height and is equal to h.

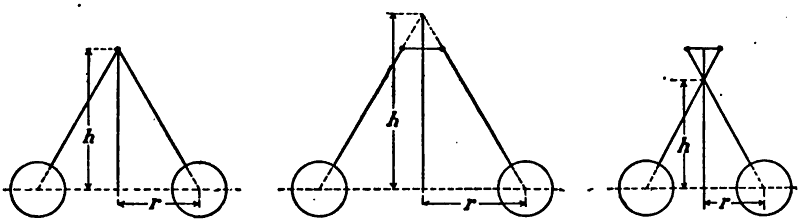


Fig. 35.

If the balls revolve faster, the centrifugal force increases, r becomes greater and h diminishes. We know that centrifugal force is expressed by the formula,

$$F = \frac{Wv^2}{g\,r}$$

Then centrifugal force varies inversely as the radius.

While the pendulum is revolving, centrifugal force acts horizontally outward and tends to make the balls fly from the center; gravity tends to make the balls drop downward. In order that the balls shall revolve at a certain height, the moments of these two forces about the center must be equal, or the weight of the balls multiplied by their distance from the center must equal the centrifugal force multiplied by the height, or.

$$W \times r = F \times h$$

from which, $$\frac{h}{r} = \frac{W}{F}$$

or, $$\frac{h}{r} = \frac{W}{\frac{Wv^2}{g\,r}} = \frac{g\,r}{v^2}$$

from which, $$h = \frac{g\,r^2}{v^2}$$

Now we know that the linear velocity of a point revolving in the circumference of a circle is expressed as $2 \pi r N$ feet per second.

Then, $$h = \frac{g r^2}{v^2} = \frac{g r^2}{4\pi^2 r^2 N^2} = \frac{g}{4\pi^2 N^2}$$

Since we know the values of g and π we can write the formula,

$$h = \frac{32.16}{4 \times 3.1416^2 \times N^2} = \frac{.8146}{N^2} \text{ feet per second, or}$$

$$h = \frac{9.775}{N^2} \text{ inches per second.}$$

If N is the number of revolutions per minute, since $60^2 = 3600$,

$$h = \frac{2{,}932.56}{N^2} \text{ feet}$$

$$h = \frac{35{,}190.7}{N^2} \text{ inches}$$

From the above formula, we see that the height is independent of the weight of the balls or the length of the rod; it depends upon the number of revolutions. The height varies inversely as the square of the number of revolutions.

The ordinary pendulum governor is not isochronous; that is, it does not revolve at a uniform speed in all positions; the speed changes as the angle between the arms and the spindle changes.

The early form consisted of two heavy balls suspended by links from a pin connection in a vertical spindle, as shown in Figs. 36 and 37. The spindle is caused to revolve by belting or gearing from the main shaft, so that as the speed increases, centrifugal force causes the balls to revolve in a circle of larger and larger diameter. The change of position of these balls can be made to affect the controlling valves so that the admission or the throttling shall vary with their position. With this governor it is evident that for a given speed of the engine there is but one position possible for the governor; consequently one amount of throttling or one point of cut-off, as the case may be. If the load varies, the speed of the engine will change. This causes the position of the governor balls to be changed slightly, thus altering the pressure

But in order that the pressure or cut-off shall remain changed, the governor balls must stay in their new position. That is to say, the speed of the engine must be slightly changed. Thus with the old ball governors there was a slightly different speed for each load. This condition has been greatly improved by various modifications until now such governors give excellent regulation.

While the engine is running with a light load, the valve controlled by the governor will be open just enough to admit steam at a pressure that will keep the engine running at a given speed. Now, if the engine is heavily loaded, the throttle valve must be wide open. The change of opening is obtained by a variation in the height of the governor, which is caused by a change of speed. Thus we see that the governor can control the speed only within certain limits which are not far apart. The difference in the extreme heights of the governor must be sufficient to open the throttle its entire range. In most well-designed engines the speed will not vary more than 4 per cent; that is, 2 per cent above or below the mean speed.

From the formula $h = \frac{35,190.7}{N^2}$, we can compute the heights corresponding to given speeds as shown by the following table:

Number of Revolutions per Minute.	Height in Inches.	Variation of Height in Inches 4 per cent.
250	.563	.0225
200	.879	.035
175	1.149	.046
150	1.564	.062
125	2.252	.090
100	3.519	.140
75	6.256	.250
50	14.076	.563

In the above table the second column is found from the formula $h = \frac{35,190.7}{N^2}$. The third column is the variation in height for a speed variation of 4 per cent or 2 per cent either above or below the mean.

From the table we see that for a considerable variation of speed there is but slight variation in the height of the governor. Also for high speeds the height of the governor is so small that it would be difficult to construct it. The slight variation in height is too small to control the cut-off or throttling mechanism throughout the entire range.

Other disadvantages of the fly-ball governor are as follows: it is apparent that the valves must be controlled by the weight of the governor balls. In large engines this requires very heavy balls in order to quickly overcome the resistance of the valves. But these large balls have considerable inertia and will therefore be reluctant to change their speed with that of the engine. The

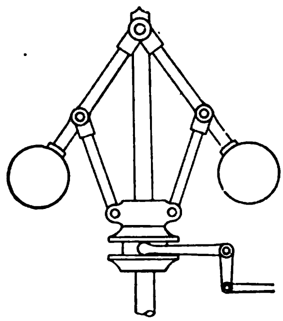


Fig. 36.

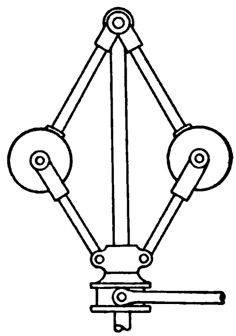


Fig. 37.

increased weight will also increase the friction in the governor joints, and the cramping action existing when the balls are driven by the spindle will increase this friction still further. All these things tend to delay the action of the governor, so that in all large engines the old-fashioned governor became sluggish. The balls had to turn slowly because they were so heavy; this was especially troublesome in high-speed engines.

To remedy these defects the weighted or Porter governor was designed. (See Fig. 38.) It has a greater height for a given speed, and the variation in height for a given variation of speed is greater. When a governor has this latter quality, that is, a great variation in height for a given variation of speed, it is said to be sensitive. By increasing this variation in height the sensitiveness is increased. Thus, if a governor running at 50 revolutions has a

variation in height of .57 inch, it is not as sensitive as one having a variation of 1 inch for the same speed.

In the weighted governor, the weight is formed so that the center of gravity is in the axis. It is placed on the spindle and is free to revolve. The weight adds to the weight of the balls, and thus increases the moment of the weight. It does not, however, add to the centrifugal force, and hence the moment of this force is unchanged. We may then say the weight adds effect to the weight but not to the centrifugal force, and as a consequence the height of the governor, for a given speed, is increased. If we let W equal the combined weight of the balls as before, and W′ equal the added weight, the moments are,

$$(W + W') \times r = F\,h$$

$$(W + W') \times r = \frac{W\,v^2}{g\,r} \times h$$

$$h = (W + W')\,r \times \frac{g\,r}{W\,v^2}$$

$$= \frac{(W + W')\,r^2\,g}{W \times 4\,\pi^2\,r^2\,N^2}$$

$$= \frac{(W \times W')}{W} \times \frac{g}{4\,\pi^2\,N^2}$$

We know that $\frac{g}{4\,\pi^2\,N^2} = \frac{.8146}{N^2}$.

Then $h = \left\{\frac{W + W'}{W}\right\} \times \frac{.8146}{N^2}$.

Hence the height of a weighted governor is equal to the height of a simple pendulum governor multiplied by

$$\left\{\frac{W + W'}{W}\right\} \text{ or } \left\{1 + \frac{W'}{W}\right\}.$$

For instance, if the height of a simple pendulum is 10 inches, and the weight of the balls equal to the added weight, the height will be,

$$h = \left\{\frac{1+1}{1}\right\} \times 10$$

$$= 2 \times 10$$

Thus we see that if a weight equal to the combined weight of the balls is added, the height of the governor will be doubled.

We know that if the balls fall, the cut-off comes later. If the belt driving the governor slips off or breaks, the balls will drop, and, making the cut-off later, will allow the engine to "run away." To diminish this danger many governors are provided with some kind of safety stop, which closes the valve when the governor loses its normal action. Usually a trip is provided which the governor does not touch in its normal positions, but which will be released if the balls drop down below a certain point.

In another arrangement, instead of a weight, a strong spring is used, and this makes it possible to put the governor in any position.

Spring Governors. In many cases a spring is used in place of the weight. This type of governor is used frequently on throttling engines; it consists of a pendulum governor with springs added to counteract the centrifugal force of the balls. Thus the height and sensitiveness are increased. Fig. 39 shows the exterior view of a Waters governor, and Fig. 40 the same governor having the safety stop. In this governor the weights are always in the same plane, the variation in height being due to the action of the bell crank levers connecting the balls and spindle. When the balls move outward the spindle moves downward and tends to close the valve. The governor balls revolve by means of a belt and bevel gears. The valve and seat are shown in section in Fig. 41. The valve is a hollow cylinder with three ports, by means of which steam enters the valve. The seat is made in four parts, that is, there are four edges that the steam passes as it enters the valve. The valve being cylindrical and having steam on both sides is balanced, and because of the many openings only a small travel is necessary.

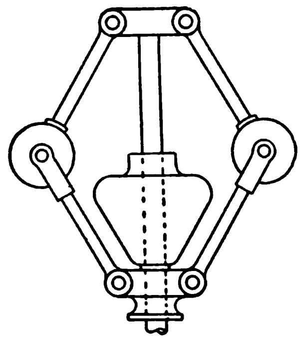


Fig. 38.

Shaft Governors. Usually some form of pendulum governor

is used for throttling engines. For governing an engine by varying the point of cut-off, shaft governors are generally used; however, Corliss engines and some others use pendulum governors for this purpose. Cut-off governors are called shaft governors because they are placed on the main shaft; they are made in many forms, but the essential features of all are the same. Two pivoted masses or weights are arranged symmetrically on opposite sides of the shaft, and their tendency to fly outward when the speed increases is resisted by springs. The outward motion of the weights closes the admission valve earlier, and the inward motion closes it later. This change is effected by altering the position of the eccentric, either by changing the eccentricity or the angular advance.

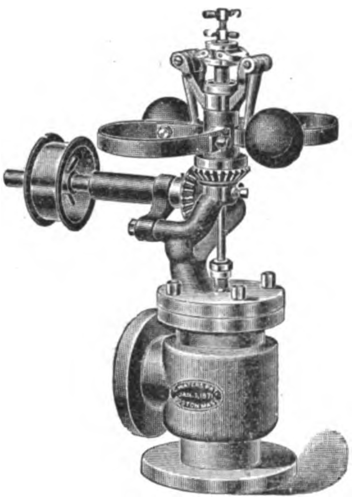


Fig. 39.

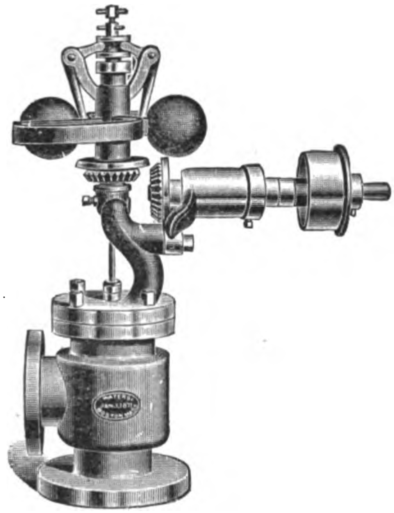


Fig. 40.

Shaft governors are made in a great variety of ways, no two types being exactly alike. If the principles of a few types are understood, it is easy to understand others. The following illustrates two common methods of shifting the eccentric.

Buckeye Engine Governor. The valve of the Buckeye engine is hollow and of the slide valve type. The cut-off valve is inside. The change of cut-off is due to the alteration of the angular advance. The arrangement of the parts which effect the change of angular advance is shown in Fig. 42. A wheel which

contains and supports the various parts of the governor is keyed to the shaft. Two arms, having weights A A at the ends, are pivoted to the arms of the wheel at *b b*. The ends having the weights are connected to the collar on the loose eccentric C by means of rods B B.

When the weights move to the position indicated by the dotted lines, the eccentric is turned on the shaft about a quarter of a revolution in the direction in which the engine runs. That is, the eccentric is advanced or the angular advance is increased. Now we know that if the angular advance is increased, cut-off occurs earlier. This is shown by the table on page 12 of "Valve Gears." If the engine had a single plain slide valve the variation of the angular advance would produce too great a variation of lead; but as this engine has a separate valve for cut-off, admission is not altered by the cut-off valve.

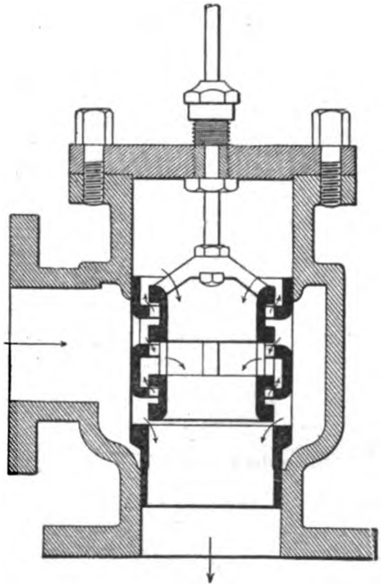


Fig. 41.

The springs F F balance the centrifugal force of the weights. The weights A A are varied to suit the speed; the tension on the springs is altered by means of the screws *c c*. Auxiliary springs are added in order to obtain the exactness of regulation necessary for electric lighting. These springs tend to throw the arms outward, but act only during the inner half of this movement.

The Straight-Line Engine Governor. Fig. 43 shows the governor of the straight-line engine. It has but one ball, B, which is linked to the spring S and to the plate D E, on which is the eccentric C. When the ball flies outward in the direction indicated by the arrow F, the eccentric is shifted about the pivot O; the links moving in the direction of arrow H. The ball is heavy and at a considerable distance from the center; hence it has a great centrifugal force, and the spring must be stiff.

The governor of the Buckeye engine alters the cut-off by changing the *angular advance.* The straight-line engine governor changes the *travel* of the valve. Shaft governors which alter the cut-off by changing the valve travel are very common.

LUBRICATION.

If two pieces of cast iron, just as they come from the foundry, are rubbed together, they will not slide over each other easily, because of little projections. If this same iron is filed or planed,

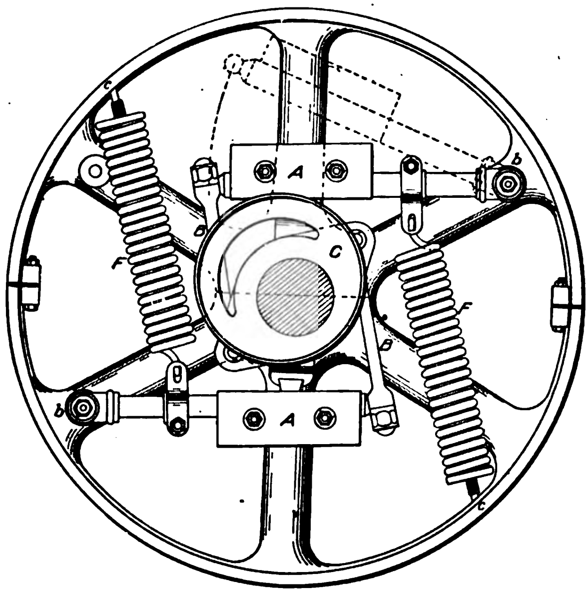


Fig. 42.

the pieces will slide much more easily. This is because the rough places have been smoothed, or filled up with dust. If now we put some engine oil on the pieces, they will slide very easily. This is because the more minute depressions have been filled up and the whole surface is made comparatively smooth. No matter how carefully we might plane and polish any surface, a microscope would show that it was still a little rough.

One cause of loss of power in the steam engine is friction. In all engines there are so many moving parts that it is of great importance that friction should be reduced as much as possible. This is

done by making the surfaces in contact smooth and of ample size; also making them of different metals and using oils or other lubricants. The effect of the lubricant is to interpose a thin film between the surfaces. This prevents their coming into actual contact. If the oil is too thin or the pressure too great, the lubricant is squeezed out and the metal surfaces come in contact.

Thus we see that there are certain qualities which a lubricant must have. They are as follows:

The lubricant must be sufficiently fluid, so that it will not itself make the bearing run hard.

It must not be too fluid or it will be squeezed out from between the bearing surfaces. If this happens, the bearing will

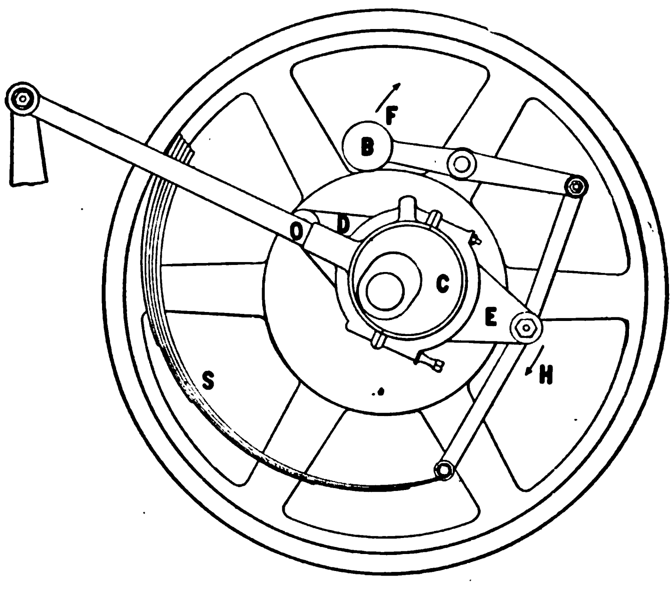


Fig. 43.

immediately begin to heat and cut. The heating will tighten the bearing, and will thus increase the pressure and the cutting.

It must not gum or dry when exposed to the air.

It must not be easily decomposed by the heat generated. If it should be decomposed, it might form substances which would be injurious to the bearings.

It must not take fire easily.

It must contain no acid, and should form no acid in decomposing, as acids corrode the bearings.

Both *mineral* and *animal* oils are used as lubricants. Formerly animal oils were used entirely, but they were likely to decompose at high temperatures and form acids. It is important in using high-pressure steam to have "high-test oils," that is, oils which will not decompose or volatilize at the temperature of the steam. It was the difficulty of getting such oils which made great trouble when superheated steam was first used. Mineral oils will stand these temperatures very readily, and even if they do decompose, they form no acids.

The Liquid Lubricants, whether of animal, vegetable or mineral origin, may be used for ordinary bearings, but for valves and pistons heavy mineral oils only are suitable.

Solid Lubricants. *Graphite* is used as a lubricant. It is well adapted for heavy pressures when mixed with certain oils. It is especially valuable for heavy pressures and low velocities.

Metalline is a solid compound, containing graphite. It is made in the form of solid cylinders, which are fitted to holes drilled into the surface of the bearing. When a bearing is thus fitted no other lubricant is necessary.

Soapstone in the form of powder and mixed with oil or fat is sometimes used as a lubricant. Soap mixed with graphite or soapstone is often used where wood is in contact with wood or iron.

A preparation called *Fiber Graphite* is used for self-lubricating bearings. It is made of finely divided graphite mixed with fibers of wood. It is pressed in molds and afterward fitted to bearings.

For great pressure at slow speed, graphite, lard, tallow and other solid lubricants are suitable. If the pressure is great and the speed high, castor, sperm and heavy mineral oils are used.

For low pressure and high speed, olive, sperm, rape and refined petroleum give satisfaction.

In ordinary machinery, heavy mineral and vegetable oils and lard oil are good. The relative value of various lubricants depends upon the prevailing conditions. Oil that is suitable for one place might not flow freely enough for another.

The quality of oil is of great importance. In many branches

of industry it is imperative that the machinery run as perfectly as possible. On this account and because of the high cost of machinery, only first-class oil should be used. The cylinder oil especially, should be high grade, because the valves, piston and piston rods are the most delicate parts of the engine.

Engines are lubricated by means of oil cups and wipers placed on the bearings wherever required. They are made in many forms, dependent upon the manufacturers. Commonly the oil cup is made with a tube extending up through the oil. A piece of lampwick or worsted leads from the oil in the cup to the tube. Capillary attraction causes the oil to flow continuously and drip down the tube. When not in use, the lampwick should be withdrawn.

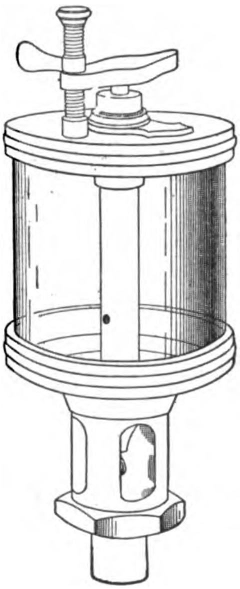


Fig. 44.

The needle oil cup differs from the capillary oil cup in that a small wire or needle extends through the tube and oil; one end rests on the journal to be lubricated. The needle should fit the tube closely, so that when the machinery is at rest no oil will flow. When revolving, the shaft gives the needle a wabbling motion which makes the oil flow. To increase the supply, a smaller needle is used.

The oil cup shown in Fig. 44 is simple and economical. The opening of the valve is regulated by an adjustable stop. The oil may be seen as it flows drop by drop. The cylindrical portion is made of glass, so that the engineer can see how much oil there is in the cup without opening it.

A form of wiper crank pin oiler is shown in Fig. 45. The oil cup is attached to a bracket. The oil drops from the cup into a sheet of wicking or wire cloth and is removed at each revolution of the crank pin by means of the cup which is attached to the end of the connecting rod.

Fig. 46 shows a centrifugal oiling device. The oil flows from the oil cup through the tube to the small hole in the crank

pin by centrifugal force. It reaches the bearing surface by means of another small hole.

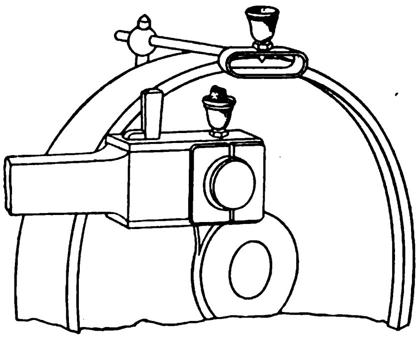


Fig. 45.

In oiling the valve chest and cylinder the lubricant must be introduced against the pressure of the steam. This can be done in several ways, in each of which it is introduced into the steam before it reaches the valve chest and is carried to the surfaces to be lubricated.

The oil can be forced into the steam pipe by a small hand pump or in large engines by an attachment from the engine itself. The supply of oil is, of course, intermittent if the pump is driven by hand, but continuous and economical if driven by the engine.

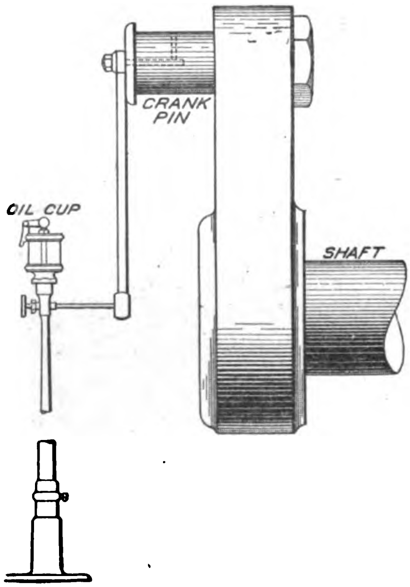


Fig. 46.

Sight Feed Lubricators. The most common device for feeding oil to the cylinder is that which introduces the oil drop by drop into the steam when it is in the steam pipe or steam chest. The oil becomes vaporized and lubricates all the internal surfaces of the engine.

Fig. 47 shows the section of a sight feed lubricator. The reservoir O is filled with oil. The pipe B, which connects with the steam pipe, is partly filled with condensed steam, which flows down the small curved pipe E to the bottom of the chamber O. A small portion of the oil is thus displaced and flows from the top of the reservoir O down the tube F, by the

regulating valve D, and up through the glass tube S, which is filled with water. It enters the main steam pipe through the connection A. The gage glass G indicates the height of water in the chamber O. To fill the lubricator, close the regulating valve D and the valve in pipe B; the oil chamber can then be drained and filled. If the glass S becomes clogged it may be cleaned by shutting valve D and opening the small valve H. This will allow steam to blow through the glass. After cleaning close valve H and allow glass S to become filled with water before opening the feed valve. The amount of oil fed to the cylinder can be regulated by opening D (Fig. 47) the proper amount. The exact quantity of oil necessary for the engine is not easily determined. For ordinary sizes it is between one drop in two minutes and two drops per minute.

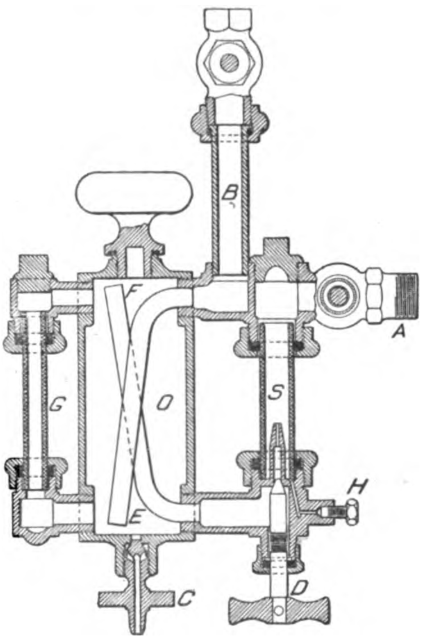


Fig. 47.

Graphite is an excellent lubricant and can be introduced into the cylinder dry or mixed with some heavy grease. It has been used extensively because of the trouble which cylinder oil gives in the exhaust and in the boilers of condensing engines.

In slow-speed engines it is not hard to attend to the oiling; all the parts are moving slowly and can be readily examined and oiled. Many high-speed engines run so fast that it is impossible to examine the various parts, and special means must be provided for lubricating. It is specially important in high-speed engines that there should be no heating. High-speed engines are generally used for electric lighting, and it is absolutely essential that they be kept running at the required speed to avoid flickering lights. Thus, while there is greater liability to heating in high-

speed engines, there is also much greater loss in case heating compels the stopping of an engine.

In order to avoid the danger of forgetting to oil a bearing of a high-speed engine, it is customary to have all the bearings oiled from one place. All the oil is supplied to one reservoir and from this reservoir pipes lead to all bearings. If this is not done, large oil cups are supplied, as a rule, so that oiling need not be attended to as frequently.

In some high-speed engines the moving parts are enclosed and the crank runs in a bath of oil. This secures certain oiling and is very effective. All the bearings may be inside this crank case, so that all are oiled in this way. It is impossible for a careless engineer to overlook one point and so endanger the whole engine.

STEAM TURBINE.

The very earliest records of the steam engine describe a form of steam turbine. It consisted of a hollow sphere, as shown in Fig. 48, mounted on trunnions, through which steam was admitted to the interior. This steam escaped through pipes bent tangentially to the equator line of the sphere. The force of the escaping steam reacted upon the sphere, causing it to revolve on its trunnions. Many centuries later, in 1629, Branca, an Italian, invented a rotary engine (Fig. 49), in which a jet of steam struck the vanes of a wheel, and thus forced it around in much the same way that a jet of water acts on a Pelton water-wheel. These engines were of little, if any, practical value, and used an immense quantity of steam.

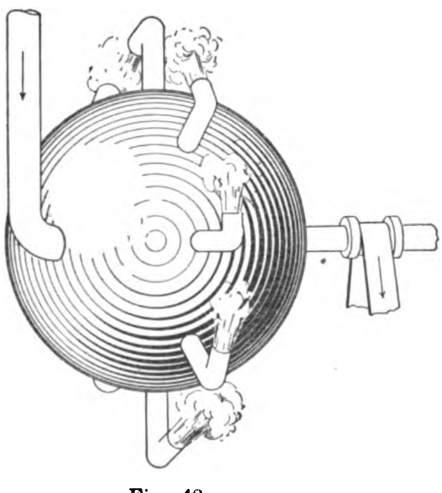


Fig. 48.

In 1705 the reciprocating engine was introduced, and by means of Watt's inventions became so efficient that the development of the rotary engine was out of the question. It will be remembered that Watt introduced the expansive use of steam in the reciprocating engine, which at this time could not be accomplished in the rotary engine, and until within the last few years practically nothing was done to develop the turbine.

Since the days of Watt there has been but one important thermodynamic improvement in the reciprocating engine; namely,

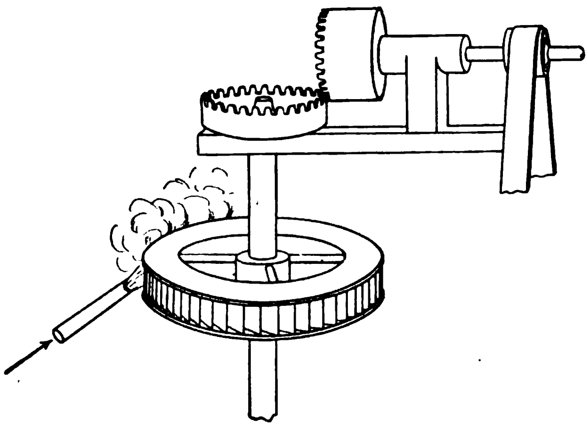


Fig. 49.

the introduction of compound expansion. All other improvements have been in the nature of mechanical devices, and it seems reasonable to suppose that the greatest developments of the future may possibly be in the production of some type other than the reciprocating engine.

In 1883 De Laval invented a successful turbine for running a cream separator, and a short time later Parsons introduced another. Both of these engines employed the expansive force of steam, but each derived this force in a different way.

Since 1883 the development of the turbine has been very rapid. The first engine introduced by De Leval, although far ahead of the earlier forms, was still very wasteful of steam; but now such improvements have been made that their steam consumption compares very favorably with the consumption of good reciprocating engines.

A modern turbine is a tremendously high-speed engine. It does not derive its power from the static force of steam expanding behind a piston, as in the reciprocating engine, but in this case the expanding steam produces kinetic energy of the steam particles. These particles receive a high velocity by virtue of the expansion, and, acting upon the vanes of a wheel, force it around at a high speed of rotation in some such manner as a stream of water rotates a water-wheel.

In the reciprocating engine the expansion produces a force which presses on the piston. In the rotary engine the expansion produces velocity in a jet of steam. This is the fundamental difference between the two forms.

The essential principles of water turbines are equally true of steam turbines. The jet must strike the vanes without a sudden shock, and must leave them in another direction without any sharp deflections. For maximum efficiency the De Laval engine should have a jet velocity equal to one-half the linear velocity of a point on the wheel-rim; for the Parsons these velocities should be equal. If the velocity of steam is 8,000 feet per second, it is easily seen that even one-half of this would cause too great a speed of rotation for safety. It would be difficult to build a wheel that would be strong enough to withstand the centrifugal force at this high speed. It becomes necessary, therefore, to reduce the speed to the limits of safety, and run under a slightly less efficiency.

At such high speed the shaft and wheel should be perfectly balanced, in order that its center of gravity may exactly coincide with the axis of rotation. In practice it has been found impossible to balance the shaft perfectly; and in order that it may revolve about its center of gravity, various means are adopted to overcome the rigidity of an ordinary shaft and bearing. This makes high speed of rotation possible without any apparent vibration.

De Laval Turbine. The De Laval turbine shown in Fig. 50 consists of a wheel with suitably shaped buckets, against which a jet of steam is directed. The buckets are on the rim of the wheel and are surrounded by a casing B, which prevents the escape of the steam until it has done its work. A piece of this casing is cut away at A in order to show the buckets. The steam

from the nozzle strikes these buckets and is deflected. Thus by the impact of the jet and the reaction due to its deflection, the wheel is caused to revolve at a high speed.

There are usually four nozzles that supply steam to the turbine, one of which is shown in section at D. These nozzles are small at the throat and diverge outward. By making them of the right length and with the proper amount of divergence, the steam can be expanded from the pressure of admission to the

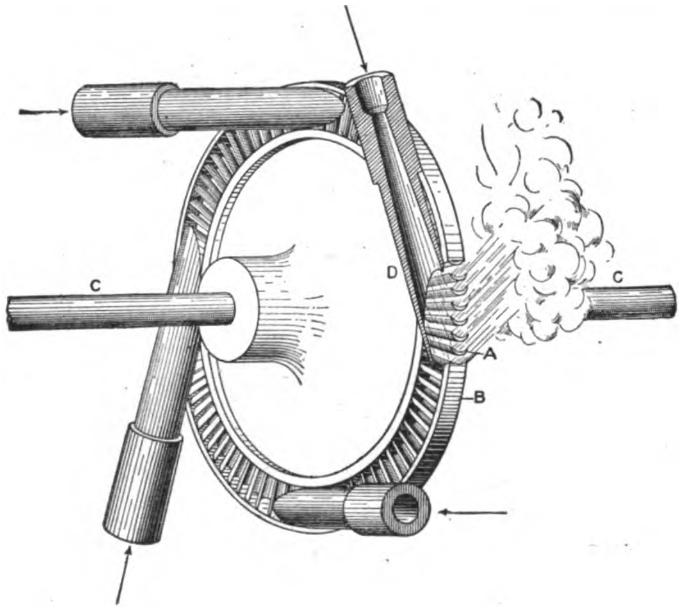


Fig. 50.

pressure of the condenser. Complete expansion is obtained in this diverging nozzle, and the steam leaves it at the exhaust pressure. The steam then works only by virtue of its high velocity.

This turbine has a long, flexible shaft C which can deflect enough to make up for any eccentricity of the center of gravity of the shaft, and thus allow the shaft to revolve about its center of gravity and still have rigid bearings at the end.

Admission is regulated by a throttle-valve, controlled by a fly ball governor.

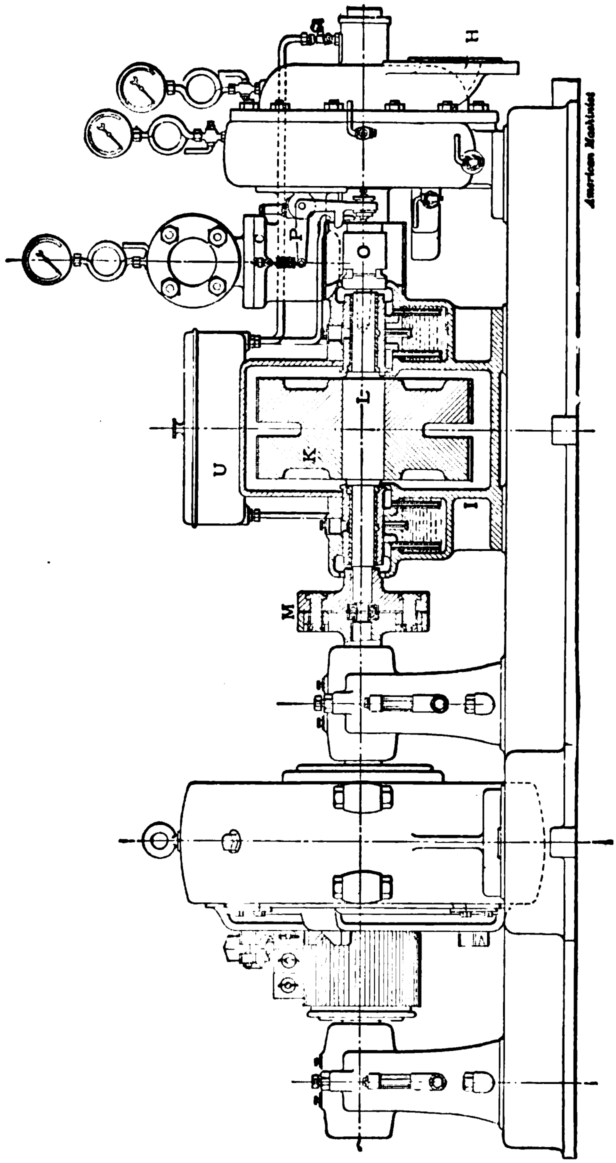


Fig. 51.

Fig. 51 shows a De Laval turbine connected with a generator.

The Parsons Turbine. Fig. 52 is a longitudinal section of a Westinghouse-Parsons turbine. Steam enters the chamber A and passes through the turbine vanes to the exhaust chamber B. The vanes are arranged as shown in Fig. 53, and consist of alternate sets, one stationary, the next movable. The steam strikes one and is deflected to the next; thus the action and the reaction occurring in rapid succession cause the movable sets, which are fixed to the shaft, to rotate at a high speed. As the steam passes the different sets of blades, the volume of the passages increases to correspond with the expansion of the steam. In the De Laval the steam was expanded entirely before reaching the wheel, but in the Parsons the expansion is accomplished in the engine itself. As the steam enters the chamber A (see Fig. 52) it presses on the turbine vanes and it also presses equally and in the opposite direction on C, which is really a piston fixed to the shaft. Thus we see that the pressure to the right is equal to the pressure to the left, and there is no end pressure on the bearing of the shaft. C_1 and C_2 balance the steam pressure in the chambers E and G. At H is a bearing which serves to maintain a correct adjustment of the balance pistons C. There is probably some escape of steam past these balancing pistons, but it is small. The exhaust steam at B presses the turbine toward the left, and would cause an end pressure on the bearing were it not that the pipe K opens a communication between the exhaust chamber B and the back of the balancing pistons, which makes the pressure equal at both ends.

The bearing consists of a gun-metal sleeve surrounded by three concentric tubes. There is a small clearance between these tubes which fills with oil and permits the bearing to run slightly eccentric to counteract any lack of balance in the shaft. Thus the shaft may revolve about its center of gravity, and this oil bearing serves the same purpose as the De Laval flexible shaft.

At P is shown a by-pass valve by means of which live steam may be admitted to the space E, if desirable. Of course this reduces one stage of the expansion, with a corresponding loss of economy, but will increase the power of the turbine. If the condenser fails on a condensing turbine it may still be run at full load by opening the by-pass valve.

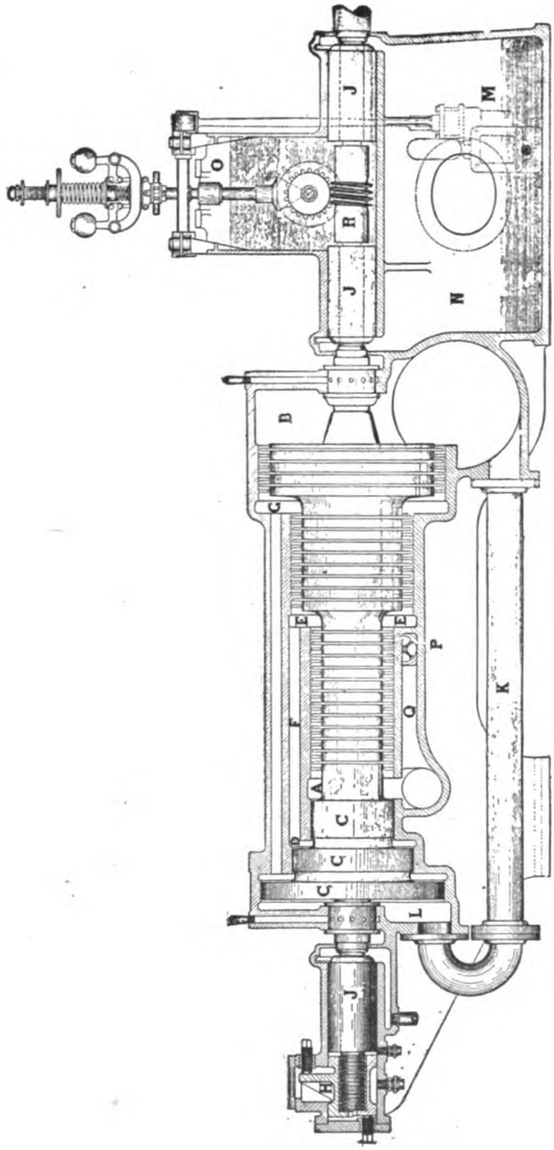


Fig. 52.

Steam is admitted to this turbine in puffs through a reciprocating valve. A fly-ball governor regulates the admission, which is always at boiler pressure.

For electric generators the turbine has many advantages, among them high speed and direct connection. They have small foundations and take up little space; there is slight loss from friction and few parts. Where slow speed is desired a reciprocating engine is probably the best.

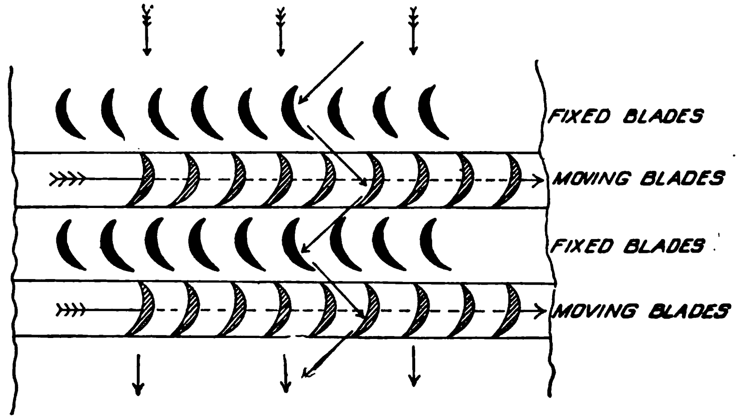
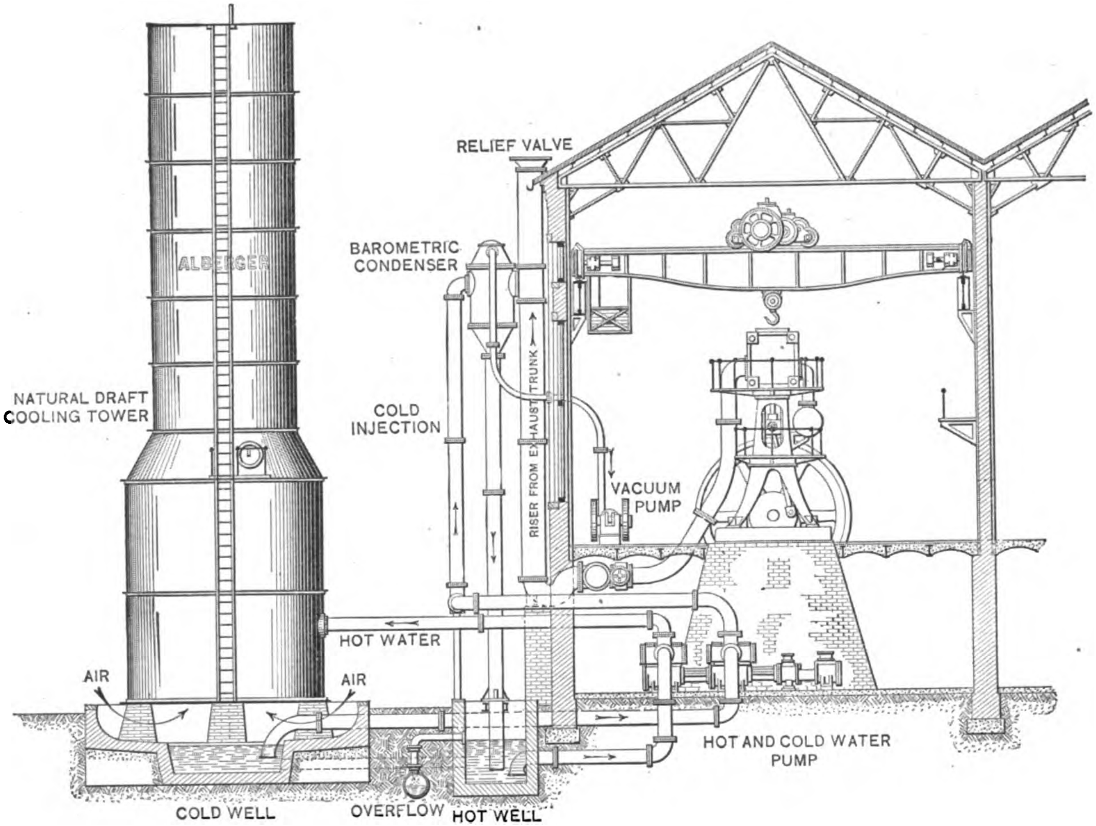


Fig. 53.



BAROMETRIC CONDENSER AND NATURAL DRAFT COOLING TOWER.
Alberger Condenser Co.

THE STEAM ENGINE

PART II

ACTION OF HEAT.

There are several types of heat engines, such as the steam engine, the gas engine, the hot-air engine, etc., each one of which derives its motive power from the heat contained in steam, gas, oil, hot air, or some other substance. Heat is imparted to these substances either by the combustion of fuel in a generator entirely separate from the engine, or by the combustion of a gaseous substance in the cylinder of the engine itself. In the case of the hot-air engine the heat is produced by a fire immediately beneath the engine.

Steam and hot gases have a tendency to expand, because of the heat they contain, thus producing a pressure in all directions. This pressure causes the piston of the engine to move, which allows the gas to expand. As the gas expands, it gives up heat, which is converted into useful work.

We shall now discuss the fundamental principles of the action of heat, and the behavior of gases and vapors, with special reference to the properties of steam and its action in the cylinder of the engine.

If a piece of iron or some other substance is placed in a fire, it becomes hotter than it was before, because heat from the fire has passed into it. If this hot substance is plunged into cold water, or is allowed to remain in a cool place, heat will pass from it, and we say that it becomes cold.

There have been many theories as to what heat is. The accepted theory of to-day is that heat is the result of motion, or we may say it is a form of kinetic energy. Heat is produced, not by the motion of the substance itself (for the hot body may be at rest), but by a rapid vibration of the minute individual particles that make up the body. These minute particles are called molecules. The faster the molecules vibrate, the greater will be their

kinetic energy and the hotter the substance will become. The hottest bodies are those that have the greatest energy of molecular vibration. A hot body can transfer its energy of vibration to another, which in turn becomes hotter than it was before, while the first body loses a part of its energy of vibration and becomes cooler. The terms hot and cold are only comparative terms; one body is hot because it contains a greater degree of heat than another; the other is cold because it contains a less degree of heat than the first. Cold, then, is simply a low degree of heat.

By temperature is meant simply the thermal condition of a body with reference to its capability of transferring heat to other bodies. If two bodies are placed in contact and the first gives more heat to the second than it receives, we say that No. 1 is hotter than No. 2. If the first receives more heat than it gives, No. 2 is hotter than No. 1. If both bodies give and receive the same amount of heat, they are of the same temperature.

According to our theory, it is evident that temperature depends upon the energy of molecular vibration. If the temperature rises, it means that the molecular vibration, and consequently the energy, increases; if the temperature falls, the energy of molecular vibration decreases. Evidently a point must finally be reached when this energy of vibration is zero and the molecules are at rest. At this temperature there is no heat and we call it the absolute zero. It is evident that this zero is very much below the zero of the ordinary scale.

In order to determine just how hot a body is, we must compare its temperature with that of some substance whose degree of heat we know. It would be impossible to keep several bodies at different degrees of heat for comparison, so we must resort to some other means. A simple method is to use some substance whose volume changes a definite amount for a definite change in temperature and always has the same volume for the same temperature. Mercury and alcohol are suitable substances and may be placed in a glass bulb, to which is connected a glass tube of small bore. All the air is drawn out of the tube, and the end is sealed so that the thermometric substance can expand or contract in a vacuum. The tube having been sealed, the bulb is placed in melting ice and the height of the mercury in the tube noted. It is then placed

in steam (or boiling water) at atmospheric pressure and the height of the column again noted. On the Fahrenheit scale the melting point is called 32°, and the boiling point 212°, and the intervening space is divided into 180 equal parts. In the Centigrade scale the melting point is called 0 and the boiling point 100°; there are 100 equal intervals between them. Thus we see that 180° F = 100° C, or 1° C = 1.8° F.

Example: What is the temperature of 50° C on the F scale?

$$50° \text{ C} = 50 \times 1.8 = 90° \text{ F above the melting point}$$
$$\text{or } 90 + 32 = 122° \text{ F above zero.}$$

In order to compare temperatures, we place the thermometer in contact with the substance whose degree of heat we wish to know and then observe the height of the liquid column in the thermometer. The height of this column depends upon the expansion of the thermometric substance and indicates the intensity of heat, or the temperature as we commonly call it. We use a thermometer to measure the *intensity* of heat, but not the *quantity* of heat.

For measuring the intensity of heat, the degree is the unit; for measuring the quantity of heat we have another unit, which is the amount of heat necessary to raise one pound of water from 61° F to 62° F. This is called the British thermal unit (B. T. U.). To raise one pound of water from 60° F to 62° F, or to raise two pounds from 60° F to 61° F, will require 2 B. T. U.

Suppose we have a small bar of iron heated to a white heat; its temperature will be high, but it will contain a relatively small quantity of heat, that is, it will not require a great many B. T. U. to raise its temperature to this high point. But suppose this same number of heat units were transferred to a ton of iron; the temperature would scarcely be changed, for the infinitely greater number of particles would have a correspondingly less number of vibrations.

Experience teaches us that when a rifle-ball strikes a target it stops, and that the energy which it possesses by virtue of its bodily motion is suddenly transformed into energy of molecular motion. Energy is indestructible; the molecular motion of the impinging body is at once increased and heat is developed. In general, whenever moving bodies are brought to rest, either sud-

denly as by impact or gradually as by friction, the kinetic energy of the moving mass is transformed into molecular kinetic energy, and we say that the bodies become heated.

We have now seen that we can transform energy of motion into heat. Also by means of suitable apparatus we can transform heat into energy of motion. Heat is the lowest form of energy, and while it is comparatively easy to transform other forms into heat, it is not as easy to change heat into the higher forms of energy. The principle of the transformation is simple enough, but if we are to have an efficient engine, we must be able to extract practically all of the heat from the working substance. This, however, is impossible, because the ordinary ranges of temperature used in practice are so far removed from the absolute zero of temperature, that with the most perfect machines we can at best recover but a fraction of the heat; the rest passing out of the engine.

The transformation is accomplished by means of a working substance which passes from the temperature of a heat generator into a refrigerator; the heat given up during the change is transformed into work. By the term refrigerator we mean the low temperature of the working substance at exhaust. The greater the temperature of the source of heat, or the lower the temperature of the refrigerator, the greater will be the amount of heat that can be abstracted and converted into useful work. If the temperature of the refrigerator could be reduced to the absolute zero, all the heat would be removed from the working substance, and the only loss would be that due to mechanical imperfections of the engine; but since the absolute zero is 461° below the zero on the Fahrenheit scale, or 493° below the freezing point, we must at best allow a relatively large amount of heat to pass out of the engine into the refrigerator without having done any work at all.

The unit of work is the foot-pound; that is, the work done in raising one pound one foot. By means of careful experiments it has been determined that for every 778 foot-pounds of work transformed into heat there is developed one B. T. U. This value 778 is known as the Mechanical Equivalent of heat. It means that one B. T. U. and 778 foot-pounds of work are mutually interchangeable.

EXPANSION OF GASES.

A perfect gas, strictly speaking, is one that cannot be liquefied; ordinarily, however, we apply the term to those gases that can be liquefied only with great difficulty, that is, under extreme pressure and a great reduction of temperature. For every perfect gas there is a definite relation between the pressure and volume, and what is known as Boyle's Law has been found to hold true, *viz.:* The pressure of a perfect gas at constant temperature varies inversely as the volume; that is, if the temperature remains constant the pressure becomes less as fast as the volume becomes greater, or conversely the volume becomes less as fast as the pressure becomes greater. Now if the pressure becomes twice as great, the volume becomes half as large, and if the volume becomes three times as great, the pressure will be only one-third as much. Hence we see that however the pressure may vary, the volume will change in such a way that the pressure multiplied by the volume will always be constant, provided the temperature remains the same.

This simple law is expressed thus:

$$P \times V = C$$

In which

$P =$ pressure in pounds per square inch (absolute)

$V =$ volume in cubic feet

$C =$ a constant which has different values for different gases.

Gases that are not easily liquefied, such as hydrogen, oxygen and air, follow this law fairly well, but those that are easily liquefied, such as steam and ammonia, do not follow it at all.

The value of C is not the same for all gases, and as no gas is, strictly speaking, a perfect gas, its value varies slightly with the temperature. For air, which is nearly a perfect gas, its value is 182.08 at 32° F.

Example.—What is the absolute pressure per square inch of one pound of air if the temperature is 32° F, and the volume 4.129 cubic feet?

$$V = 4.129$$

$$P \times V = C \quad \text{or} \quad P = \frac{C}{V} = \frac{182.08}{4.129}$$

$P = 44.1$ pounds per square inch, absolute.

Let us discuss this law by means of a diagram. Fig. 1 is drawn with O Y and O X at right angles to each other; pressures are measured to any convenient scale on O Y and volumes on O X. O C represents 12.387 cubic feet and O D represents 14.7 pounds, since P $\times$ V = 182.08. O F represents 24.774 cubic feet and O G 7.35 pounds. In like manner O L represents 6.19 cubic feet and O N 29.41 pounds. Then if we draw perpendiculars to O X at L, C and F, and perpendiculars to O Y at N, D and G, they will meet in the points M, E and H. The curve A B is drawn through these points. Then for any pressure we can find the corresponding volume or *vice versa*. The area of the rectangle O D E C equals that of the rectangle O G H F and also that of the rectangle O N M L. That this is so is readily seen from the fact that the product of the pressure and its corresponding volume is constant.

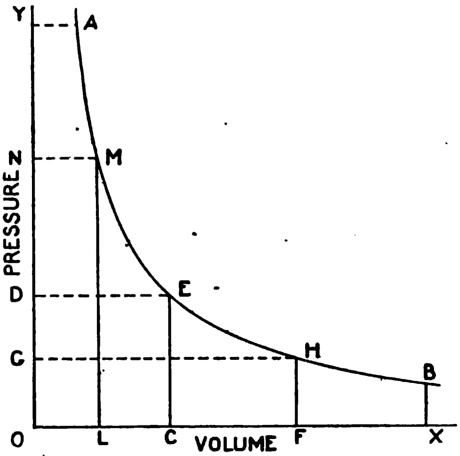


Fig. 1.

If we plot a curve using this equation we will get a rectangular hyperbola, as shown in "Steam Engine Indicators." This curve is called an isothermal curve or curve of equal temperatures.

If the volume of a perfect gas remains constant, the pressure will vary as the temperature. Or, if the pressure remains constant, the volume will vary as the temperature. This is known as the Law of Charles. From this we see that, as the temperature decreases, either the pressure or the volume will decrease a proportionate amount, and this must continue as long as there is any heat in the gas; finally a low temperature (the absolute zero) will be reached, where there is no more heat, and consequently either the pressure or the volume must be zero, provided this law holds true at such a very low temperature.

The law states that any change of pressure or volume is pro-

portional to the change in temperature, that is, the new volume is equal to the first volume plus or minus some fractional part of this volume, called the coefficient of expansion, multiplied by the change in temperature. We may express the formula thus:

$$P_1 V_1 = P V + (P V) k \times t$$
$$= P V (1 + k t).$$

Where P_1 and V_1 are the new pressure and volume
P and V are the first pressure and volume
k is the coefficient of expansion for the gas
t is the change in temperature.

From Boyle's Law we know that $P V = C$; hence we may write the above equation:

$$P_1 V_1 = C (1 + k t)$$
$$= K C (\frac{1}{k} + t)$$

From careful experiments on the expansion of air, Regnault determined the value of the coefficient k to be .003654 for Centigrade units. This value is constant between freezing point, 0, and boiling point, 100°. Substituting this value of k in our last equation, we have:

$$P_1 V_1 = C (\frac{1}{.003654} + t)$$
$$= C (273.7 + t).$$

Now if we should make the change of temperature, t, equal to — 273.7° we should have:

$$P_1 V_1 = C(273.7 - 273.7) = 0$$
$$P_1 V_1 = 0.$$

Therefore we must have reached the absolute zero at — 273.7° below the freezing point, because here $P_1 \times V_1$ has reached 0, as we have previously seen must be the case at the absolute zero. $273.7 \times 1.8 = 492.7°$ for the F scale. Therefore the absolute zero is $492.7° - 32° = 460.7°$ below zero on the Fahrenheit scale. For ordinary work 461 will be sufficiently accurate.

If we let P = absolute pressure in pounds per square inch
V = volume of one pound in cubic feet
T = absolute temperature on Fahrenheit scale,

Then, $$P V = C T.$$

C equals .3693 for air.

Example.—If one pound of air occupies 16.606 cubic feet at a pressure of 14.7 pounds per square inch, what is its temperature?

$$P \times V = C \times T$$
$$14.7 \times 16.606 = .3693 \times T$$
$$T = \frac{244.108}{.3693}$$
$$T = 661.$$

This value 661 is absolute temperature, and to find the Fahrenheit temperature 461 must be subtracted from it. Thus, $661° - 461° = 200°$ F.

Saturated Vapor. The process of converting a liquid into a vapor is known as vaporization; the product thus formed is readily condensed and therefore does not follow the laws of perfect gases at all. A dry saturated vapor is one that has just enough heat in it to keep it in the form of a vapor; if we add more heat it becomes superheated. A superheated vapor may lose a part of its heat without condensation; a saturated vapor cannot. When a saturated vapor loses a part of its heat some of it will condense and we say that the vapor is wet.

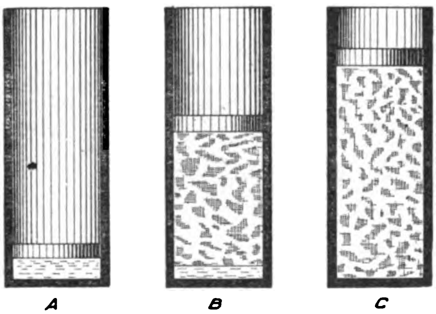


Fig. 2.

Steam is simply the vapor from water and we shall confine our discussion to this alone. Suppose we have a vertical cylinder, as shown in Fig. 2, fitted with a light piston free to move up and down, yet so constructed that it may be loaded at will. Suppose that there is one pound of water at a temperature of 32° F in the bottom of this cylinder, and that the piston rests upon its surface. Now, if we apply heat by means of a gas flame or fire, we shall notice the following effects:

First.—The temperature of the water will gradually rise until it reaches the temperature at which steam is formed. This temperature will depend upon the pressure, or the load on the

piston. If the piston is very light, we may neglect its weight and consider that there is simply the atmospheric pressure of 14.7 pounds per square inch acting on the water surface. At this pressure steam will begin to form at 212° F.

Second.— As soon as 212° F is reached, steam will begin to form and the piston will steadily rise, but no matter how hot the fire may be, the temperature of both water and steam will remain at 212° until all the water is evaporated. We had one pound of water at 32° F and at 14.7 pounds absolute pressure, and found that steam formed at a temperature of 212° F and remained at that temperature. We added 180.9 B. T. U., the heat of the liquid, to bring the water from 32° to the boiling point. To convert water at 212° into steam at 212°, we added 965.7 B. T. U. more. This quantity, known as the latent heat, or heat of vaporization, makes the total heat 1,146.6 B. T. U. If we should measure the volume carefully after all the water was evaporated, we should find that there was just 26.36 cubic feet of dry saturated steam. We had one pound of water, and therefore must have one pound of steam, for none of it could escape; hence one cubic foot will weigh $\frac{1}{26.36} = 0.03794$ pounds, which is known as the density of steam at 14.7 pounds absolute pressure or 212° F. In the Table of Properties of Saturated Steam (see page 14) all these quantities are found in the order given and at the pressure of 14.7 pounds above vacuum.

Suppose now we place a weight of 85.3 pounds on the piston. The pressure is 85.3 pounds plus 14 7 pounds, or 100 pounds absolute. We shall now find that no steam will form until a temperature of 327.58° is reached. Starting with water at 32°, it will be necessary to add 297.9 B. T. U. before a temperature of 327.58° is reached, and also we must add 884.0 B. T. U. more to vaporize it, making a total heat of 1,181.9 B. T. U. Under this greater pressure the steam occupies a volume of only 4.403 cubic feet, or one cubic foot of it weighs $\frac{1}{4.403} = 0.2271$ pound.

Of course it would be impossible to determine all these different quantities by actual experiment, and at all pressures varying from vacuum to the high pressures, used in water-tube boilers.

Fortunately we are able to compute them all from equations which have been carefully determined by experiment. If saturated steam were a perfect gas, we could easily calculate all the relations of pressure, volume and temperature from the equation $PV = CT$, but steam is so far removed from the state of a perfect gas that these relations do not hold, and the true equations become very complex. The following equation proposed by Rankine is one of the simplest, and gives fairly good results:

$$\log_{10} P = A - \frac{B}{T} - \frac{C}{T^2}$$

in which P = pressure in pounds per square inch above vacuum.
$A = 6.1007$
$B = 2{,}732$
$C = 396{,}945$
T = absolute temperature in Fahrenheit degrees.

By the aid of such equations as this, all the different quantities found in the steam tables may be calculated. These equations are based on careful experiments and give very satisfactory results.

Steam Tables. We have already seen that any change in the temperature of saturated steam produces a change of pressure, and that every change of pressure corresponds to a certain change in temperature. There are several properties of saturated steam that depend upon the temperature and pressure; and the values of all these different properties when arranged for all temperatures and pressures are called Steam Tables. The following are the principal items that are found in the tables:

1. The absolute pressure in pounds per square inch; it is equal to the gage pressure plus the atmospheric pressure of 14.7 pounds.

2. The temperature of the steam, or boiling water, at the corresponding pressure.

3. The heat of the liquid; or the number of B. T. U. necessary to raise one pound of water from 32° F to the boiling point corresponding to the given pressure.

4. The heat of vaporization, or the latent heat; this is the number of B. T. U. necessary to change one pound of water, at the boiling point, into dry saturated steam at the same temperature and pressure.

5. The total heat or the number of B. T. U. necessary to change one pound of water from 32° F into steam at the given temperature or pressure. The total heat is evidently equal to the sum of the heat of the liquid and the heat of vaporization.

6. The density of the steam; that is, the weight in pounds of one cubic foot of steam at the given temperature or pressure.

7. The specific volume; or volume in cubic feet of one pound of steam at the required temperature or pressure. Evidently the specific volume is equal to $\frac{1}{\text{density}}$.

All these properties have been calculated by means of various formulas which have been deduced from the results of actual experiment. There are several formulas for the temperatures and pressures of steam; as some computers have used one and some another, there is likely to be a slight discrepancy between the tables computed by different authors. Rankin's formula, already given, is the simplest, but is not generally considered to be quite as accurate as some of the later ones; it has probably been used, however, more than any other.

The total heat may be calculated by means of the formula

$$H = 1{,}091.7 + 0.305\,(t - 32) \qquad [1]$$

in which H = total heat

t = temperature in degrees Fahrenheit.

The heat of the liquid is equal to q.

$$q = t + 0.00002\,t^2 + 0.0000003\,t^3 \qquad [2]$$

in which t = temperature in degrees Centigrade.

These constants are for use in the Centigrade system only. To calculate the heat of the liquid for any Fahrenheit temperature it is necessary to change the Fahrenheit into equivalent Centigrade degrees and then substitute in the above formula.

The formula used for the calculation of specific volume is too complex for consideration here, but the relation of pressure and volume may be approximately expressed by means of an equation of the form $P V^n = C$, in which n is an exponent, C a constant, P the absolute pressure, and V the specific volume; n is usually taken to be $\frac{17}{16}$, and C to be 475.

On pages 14 and 15 are given tables of the properties of

TABLE OF PROPERTIES OF SATURATED STEAM.

Pressure in pounds per sq. in. above vacuum.	Temperature in degrees Fahrenheit.	Heat in liquid from 32° in units.	Heat of vaporization, or latent heat in heat units	Total heat in heat units from water at 32°.	Density or weight of cubic ft. in pounds.	Volume of 1 pound in cubic feet.	Total pressure above vacuum.
1	101.99	70 0	1043.0	1113.1	0.00299	334.5	1
2	126.27	94.4	1026.1	1120.5	0.00576	173.6	2
3	141.62	109.8	1015.3	1125.1	0.00844	118.5	3
4	153.09	121.4	1007.2	1128.6	0.01107	90.31	4
5	162.34	130.7	1000.8	1131.5	0 01366	73 21	5
6	170.14	138.6	995.2	1133 8	0.01622	61 67	6
7	176.90	145.4	990.5	1135 9	0.01874	53.37	7
8	182.92	151.5	986.2	1137.7	0.02125	47.06	8
9	188.33	156.9	982.5	1139.4	0.02374	42 12	9
10	193.25	161.9	979.0	1140.9	0.02621	38.15	10
14 7	212.00	180.9	965.7	1146.6	0 03794	26.36	14.7
15	213.03	181.8	965.1	1146.9	0.03826	26.14	15
20	227.95	196 9	954.6	1151.5	0.05023	19.91	20
25	240 04	209.1	946.0	1155.1	0.06199	16.13	25
30	250.27	219.4	938.9	1158 3	0.07360	13 59	30
35	259.19	228.4	932.6	1161 0	0.08508	11.75	35
40	267.13	236.4	927.0	1163 4	0 09644	10.37	40
45	274.29	243.6	922.0	1165 6	0.1077	9 287	45
50	280.85	250.2	917.4	1167.6	0.1188	8 414	50
55	286.89	256.3	913.1	1169.4	0.1299	7.696	55
60	292.51	261.9	909.3	1171 2	0.1409	7 097	60
65	297.77	267.2	905.5	1172 7	0.1519	6 583	65
70	302.71	272.2	902.1	1174.3	0.1628	6.143	70
75	307.38	276.9	898.8	1175.7	0.1736	5.762	75
80	311.80	281.4	895.6	1177.0	0.1843	5 426	80
85	316.02	285.8	892.5	1178 3	0.1951	5 126	85
90	320.04	290.0	889.6	1179 6	0 2058	4 859	90
95	323.89	294.0	886.7	1180.7	0 2165	4.619	95
100	327.58	297.9	884.0	1181.9	0.2271	4.403	100
105	331.13	301.6	881.3	1182.9	0.2378	4.205	105
110	334.56	305.2	878.8	1184.0	0 2484	4 026	110
115	337.86	308.7	876.3	1185.0	0 2589	3 862	115
120	341.05	312 0	874.0	1186 0	0.2695	3 711	120
125	344.13	315.2	871.7	1186 9	0.2800	3 571	125
130	347.12	318.4	869.4	1187 8	0.2904	3.444	130
140	352.85	324.4	865.1	1189.5	0 3113	3 212	140
150	358.26	330.0	861.2	1191 2	0.3321	3 011	150
160	363.40	335.4	857.4	1192 8	0.3530	2 833	160
170	368.29	340.5	853.8	1194.3	0 3737	2.676	170
180	372.97	345.4	850.3	1195.7	0 3945	2 535	180
190	377.44	350.1	847.0	1197.1	0 4153	2.408	190
200	381.73	354.6	[illegible]3.8	1198.4	0 4359	2 294	200
225	391.79	365.1	836.3	1201.4	0 4876	2 051	225
250	400 99	374.7	829.5	1204.2	0.5393	1.854	250
275	409.50	383.6	823.2	1206.8	0.5913	1 691	275
300	417.42	391.9	817.4	1209.3	0 644	1 553	300
325	424.82	399.6	811.9	1211.5	0 696	1.437	325
350	431.90	406.9	806.8	1213.7	0.748	1.337	350
375	438.40	414.2	801.5	1215.7	0.800	1 250	375
400	445.15	421.4	796.3	1217 7	0.853	1.172	400
500	466.57	444.3	779.9	1224.2	1.065	.939	500

TABLE OF PROPERTIES OF SATURATED STEAM.

Temperature in degrees Fahrenheit.	Total pressure above vacuum.	Heat in liquid from 32° in units.	Heat of vaporization, or latent heat in heat units	Total heat in heat units from water at 32°.	Density or weight of cubic ft. in pounds.	Volume of one pound in cubic feet.	Temperature in degrees Fahrenheit.
32	0.089	0.	1091.7	1091.7	0.0003	3387.	32
60	0.254	28.12	1072.1	1100.2	0.0008	1234.	60
90	0.692	58.04	1051.4	1109.4	0.0021	474.6	90
120	1.683	88.10	1034.4	1118.5	0.0049	204.4	120
140	2.877	108.2	1016.4	1124.6	0.0081	123.2	140
150	3.706	118.3	1009.4	1127.7	0.0103	97.03	150
160	4.729	128.4	1002.3	1130.7	0.0130	77.14	160
170	5.98	138.5	995.3	1133.8	0.0162	61.85	170
180	7.50	148.5	988.3	1136.8	0.0200	50.01	180
190	9.33	158.6	981.3	1139.9	0.0245	40.73	190
200	11.52	168.7	974.2	1142.9	0.0299	33.40	200
210	14.12	178.8	967.2	1146.0	0.0363	27.57	210
220	17.19	188.9	960.1	1149.0	0.0435	22.98	220
225	18.91	193.9	956.7	1150.6	0.0476	20.99	225
230	20.78	198.9	953.2	1152.1	0.0521	19.20	230
235	22.80	204.0	949.6	1153.6	0.0569	17.59	235
240	24.98	209.0	946.1	1155.1	0.0619	16.14	240
245	27.33	214.1	942.6	1156.7	0.0674	14.83	245
250	29.86	219.1	939.1	1158.2	0.0733	13.65	250
255	32.57	224.1	935.6	1159.7	0.0795	12.57	255
260	35.48	229.2	932.0	1161.2	0.0862	11.60	260
265	38.60	234.2	928.6	1162.8	0.0933	10.72	265
270	41.94	239.3	925.0	1164.3	0.1008	9.918	270
275	45.51	244.3	921.5	1165.8	0.1088	9.187	275
280	49.33	249.3	918.0	1167.3	0.1173	8.521	280
285	53.39	254.4	914.5	1168.9	0.1264	7.913	285
290	57.72	259.4	911.0	1170.4	0.1359	7.356	290
295	62.33	264.4	907.4	1171.9	0.1461	6.847	295
300	67.22	269.5	903.9	1173.4	0.1567	6.380	300
305	72.42	274.5	900.5	1175.0	0.1680	5.952	305
310	77.83	279.6	896.9	1176.5	0.1799	5.558	310
315	83.77	284.8	893.2	1178.0	0.1925	5.195	315
320	89.95	290.0	889.5	1179.5	0.2058	4.861	320
325	96.48	295.2	885.9	1181.1	0.2197	4.552	325
330	103.38	300.5	882.1	1128.6	0.2343	4.267	330
335	110.66	305.7	878.4	1184.1	0.2498	4.004	335
340	118.34	310.9	874.7	1185.6	0.2660	3.760	340
345	126.43	316.1	871.1	1187.2	0.2830	3.534	345
350	134.95	321.4	867.3	1188.7	0.3008	3.324	350
355	143.91	326.6	863.6	1190.2	0.3195	3.130	355
360	153.33	331.8	859.9	1191.7	0.3391	2.949	360
365	163.22	337.1	856.2	1193.3	0.3597	2.780	365
370	173.60	342.3	852.5	1194.8	0.3812	2.623	370
375	184.49	347.5	848.8	1196.3	0.4038	2.476	375
380	195.91	352.8	845.0	1197.8	0.4276	2.338	380
385	207.87	358.0	841.4	1199.4	0.4521	2.212	385
390	220.39	363.2	837.7	1200.9	0.4780	2.092	390
395	233.50	368.4	834.0	1202.4	0.5051	1.980	395
400	247.21	373.7	830.2	1203.9	0.5336	1.874	400
405	261.55	378.9	826.6	1205.5	0.5633	1.775	405
410	276.54	384.1	822.9	1207.0	0.5945	1.682	410
415	292.21	389.4	819.1	1208.5	0.6270	1.595	415
420	308.57	394.6	815.4	1210.0	0.6610	1.512	420
425	325.65	399.8	811.8	1211.6	0.6970	1.434	425

saturated steam, one varying with the temperatures, the other with the pressures. The tables are made out in five unit intervals; intermediate points are proportional.

Example.— Suppose we wish to find the total heat corresponding to a pressure of 112.3 pounds (gage). We first add 14.7 to the 112.3 and get 127 pounds absolute pressure. The total heat of 1 pound of steam at 125 pounds pressure is 1,186.9. The total heat at 130 pounds pressure is 1,187.8.

Difference for 5 pounds = 1,187.8 — 1,186.9 = .9
Difference for 1 pound = .9 ÷ 5 = .18
Difference for 2 pounds = 2 × .18 = .36

The total heat for 127 pounds is:

Total heat at 125 pounds = 1,186.9
Difference for 2 pounds = .36
Total heat at 127 pounds = 1,187.26

This method is called interpolation, and in many complete tables the differences for the intervals are given to facilitate the work.

If steam tables are not at hand, there are several approximate formulas that may be used for rough calculations and estimates, but it must be borne in mind that results obtained by the use of these equations are not strictly accurate, and should not be used if the regular tables can be had.

Probably the relation of temperature and pressure will be most frequently needed. If the gage pressure is between 20 pounds and 100 pounds

$$t = 14\sqrt{p} + 198 \text{ approximately} \qquad [3]$$

where t = temperature in degrees Fahrenheit
p = gage pressure in pounds per square inch.

For pressures over 100 pounds per square inch (gage) we must modify the equation thus:

$$t = 14\sqrt{p} + 198 - \left(\frac{p - 100}{11}\right) \qquad [4]$$

These equations will cover a range of pressures from 20 to 340 pounds, and give an error of less than $1\frac{1}{2}°$ in nearly all cases. From 35 pounds per square inch to 100 pounds the error is generally less than one-half of one degree.

For pressures below 20 pounds use the constant 196 instead of 198.

The latent heat may be approximately expressed by the formula,

$$l = 1{,}114 - .7\,t \qquad [5]$$

in which l = latent heat

t = temperature in degrees F.

This formula gives very good results for temperatures less than 320°, corresponding to a gage pressure of about 75 pounds. Above this pressure the formula gives slightly larger results than are found in the steam tables. At 250 pounds (gage) the formula gives 829.8 and the steam tables 825.8, so that the error will not be large in any case.

We have defined a B. T. U. as the amount of heat necessary to raise one pound of water from 61° F to 62° F. The specific heat of water is nearly constant over ordinary ranges of temperature and at 400° we find the heat of the liquid from the tables to be 373.7 B. T. U. By definition, the heat of the liquid at 32° is zero, so that a rise of 368° in temperature requires 373.7 B. T. U. If we consider the heat of the liquid proportional to the rise in temperature, our error will be 5.7 units of heat in 400°. At lower temperatures the error is much smaller, so that we may express the heat of the liquid approximately by the formula,

$$h = t - 32 \qquad [6]$$

in which h = the heat of the liquid

t = temperature in degrees as before.

The total heat is equal to the sum of the last two, or

$$H = h + l \qquad [7]$$

The relation of pressure and volume of steam may be approximately expressed by the equation

$$P\,V^n = C, \text{ or } V^n = \frac{C}{P}$$

In which P = absolute pressure

V = specific volume

n = an exponent = $\frac{17}{16}$

C = a constant = 475 nearly.

We may then write the equation

$$V^{\frac{17}{16}} = \frac{475}{P} \text{ or } V = \sqrt[\frac{17}{16}]{\frac{475}{P}} \qquad [8]$$

This is called the equation of constant steam weight; it may be solved by the aid of logarithms.

The density can of course be determined from the specific volume.

Let us apply these approximate formulas to a specific case and see how the results compare with the actual quantities given in the steam tables. For this purpose we will suppose steam at 70.3 pounds gage pressure or 85 pounds absolute.

Equation (3) $t = 14\sqrt{p} + 198$

$= 14\sqrt{70.3} + 198 = 315.40°$

From steam tables, temperature $= 316.02°$

Equation (5) $l = 1{,}114 - .7\,t$

$l = 1{,}114 - (.7 \times 315.4) = 893.2$

From steam tables, latent heat $= 892.5$

Equation (6) $h = t - 32°$

$= 315.6 - 32 = 283.4$

From tables, heat in the liquid $= 285.8$

Equation (7) $H = h + l$

$= 285.8 + 893.2 = 1{,}179.0$

From tables, total heat $= 1{,}178.3$

Equation (8) $V = \sqrt[\frac{17}{16}]{\frac{475}{P}}$

$$V = \sqrt[\frac{17}{16}]{\frac{475}{85}} = 5.100$$

From tables, specific volume $= 5.125$.

A comparison of these results shows that these formulas cannot be used when accuracy is sought; but if only approximate results are desired they will be found satisfactory. Whenever possible the steam tables should be used in preference to any approximations.

Superheated Vapor. We have seen that a saturated vapor contains just enough heat to keep it in the form of a vapor; if it

loses heat it will condense. A superheated vapor is one that has been heated after vaporization; it can lose this extra heat before any condensation will take place. A vapor in contact with its liquid is saturated; one heated after removal from the liquid is superheated.

For saturated steam there is a fixed temperature for every pressure. If we know either the pressure or the temperature, we can find the other in the steam tables. For instance, if the gage pressure of a boiler is 60.3 pounds and we wish to know the temperature, we simply add atmospheric pressure and turn to our tables and find it to be 307° (about).

With superheated steam the case is entirely different, for there is no longer the same direct relation between the temperature and pressure. In fact, the relation between temperature and pressure of superheated steam depends upon the amount of superheating. Superheated steam at 60.3 pounds gage pressure may have a temperature considerably above 307° F. At a given pressure the temperature and volume of a given weight of superheated steam are always greater than the temperature and volume of the same weight of saturated steam. The properties of superheated steam at given pressure are not constant as is the case with saturated steam.

If superheated steam were a perfect gas, we could determine the relation of P, V and T by the equation $PV = CT$; but superheated steam is not a perfect gas, hence we must modify our equation. By experiment it has been determined that the following equation is nearly correct:

$$PV = 93.5\,T - 971\,P^{\frac{1}{4}}$$

In which P = absolute pressure in pounds per square foot
T = absolute temperature
V = volume of 1 pound in cubic feet.

THE STEAM ENGINE.

We have studied the action and formation of steam, and now we shall consider its application to the steam engine. We know that steam contains a great deal of heat, and that heat can be converted into work by allowing a working substance to pass from the high temperature of the heat generator to the lower tempera-

ture of the refrigerator, during this change giving up heat, which is transformed into work. There are several forms of heat engines, all of which convert the heat contained in some substance into work. At the present time the steam engine is the most important. When of good size and properly designed and run, it is as economical as any other heat engine, and it can be more easily controlled and regulated. We shall consider first the theoretically perfect engine and then the modifications that go to make up the steam engine of to-day.

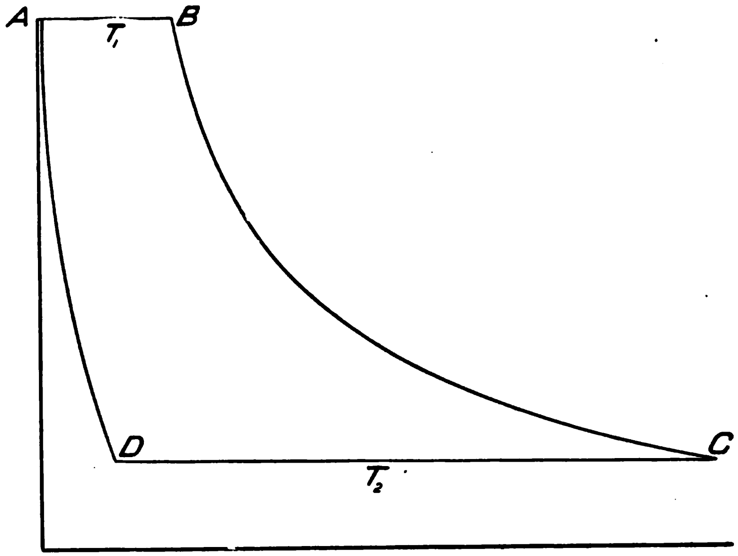


Fig. 3.

The theoretical engine (Fig. 3) is supposed to receive heat from the generator at constant temperature T_1 until communication is interrupted at B. The working substance expands to C without losing or gaining any heat from external sources until the temperature of the refrigerator is reached. The engine now rejects heat at the constant temperature T_2 of the refrigerator and then compresses the working substance without loss or gain in the quantity of heat until the temperature of the heat generator is reached. These are ideal conditions, and, if fulfilled, the efficiency of the perfect engine will depend only on the difference between the tem-

perature at which heat is received and rejected, or, in other words, it depends only upon the difference in temperature between the generator and the refrigerator.

If T_1 = absolute temperature of heat received and

T_2 = absolute temperature of heat rejected, then the thermal efficiency, E, of the engine will be represented by the formula,

$$E = \frac{T_1 - T_2}{T_1}$$

Or, in other words, the efficiency equals the absolute temperature of the heat rejected, subtracted from the absolute temperature of the heat received, and the remainder divided by the absolute temperature of the heat received.

Suppose an engine is supplied with steam at 120 pounds absolute pressure, and the exhaust is at atmospheric pressure. What is the thermal efficiency?

The absolute temperature corresponding to 120 pounds pressure is $341.05 + 461 = 802.05°$, and the absolute temperature of the exhaust is $212 + 461 = 673°$.

Then $$E = \frac{802.05 - 673}{802.05} = .16, \text{ or 16 per cent.}$$

In actual engines this efficiency cannot be realized, because the difference between the heat received and the heat rejected is not all converted into useful work. Part of it is lost by radiation, conduction, condensation, leakage and imperfect action of the valves. The cylinder walls of the theoretical engine are supposed to be made of a nonconducting material, while in the actual engine the walls are of metal, which admits of a ready interchange of heat between cylinder and steam. This action of the walls cannot be overcome, and is so important that a failure to consider its influence will lead to serious errors in computations, and no design can be made intelligently if based on the theory of the engine with nonconducting walls. The theoretical engine carries on its expansion without the loss of any heat, while in the actual engine a large amount of heat is lost by radiation. There is also a considerable loss of pressure between the boiler and engine, due to resistance of flow through pipes and passages. In a slow-speed engine with large and direct ports and valves this trouble may be

minimized. The imperfect action of valve gears may also be lessened with due care, but the action of the cylinder walls still remains to be overcome.

In the theoretical card, admission is at constant boiler pressure, cut-off is sharp and expansion complete, that is, expansion continues until the temperature falls to that of the condenser and the exhaust is at condenser pressure. The piston also sweeps the full length of the cylinder.

In the actual engine there is a considerable loss of pressure between boiler and engine, and the wire-drawing of the ports and valves tends to cause a sloping steam line. Condensation at the beginning of the stroke causes the real expansion line to fall below the theoretical, while re-evaporation causes it to rise above the theoretical toward the end of expansion. In the actual

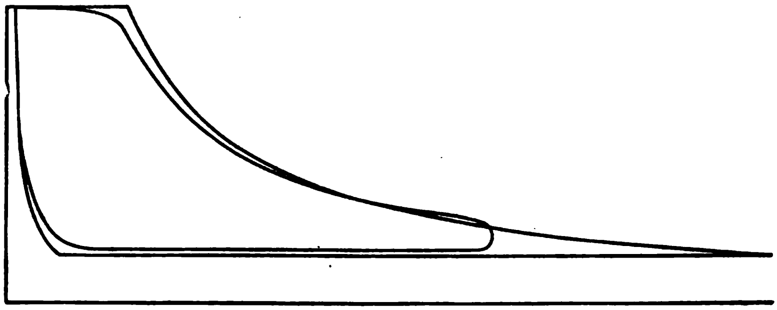
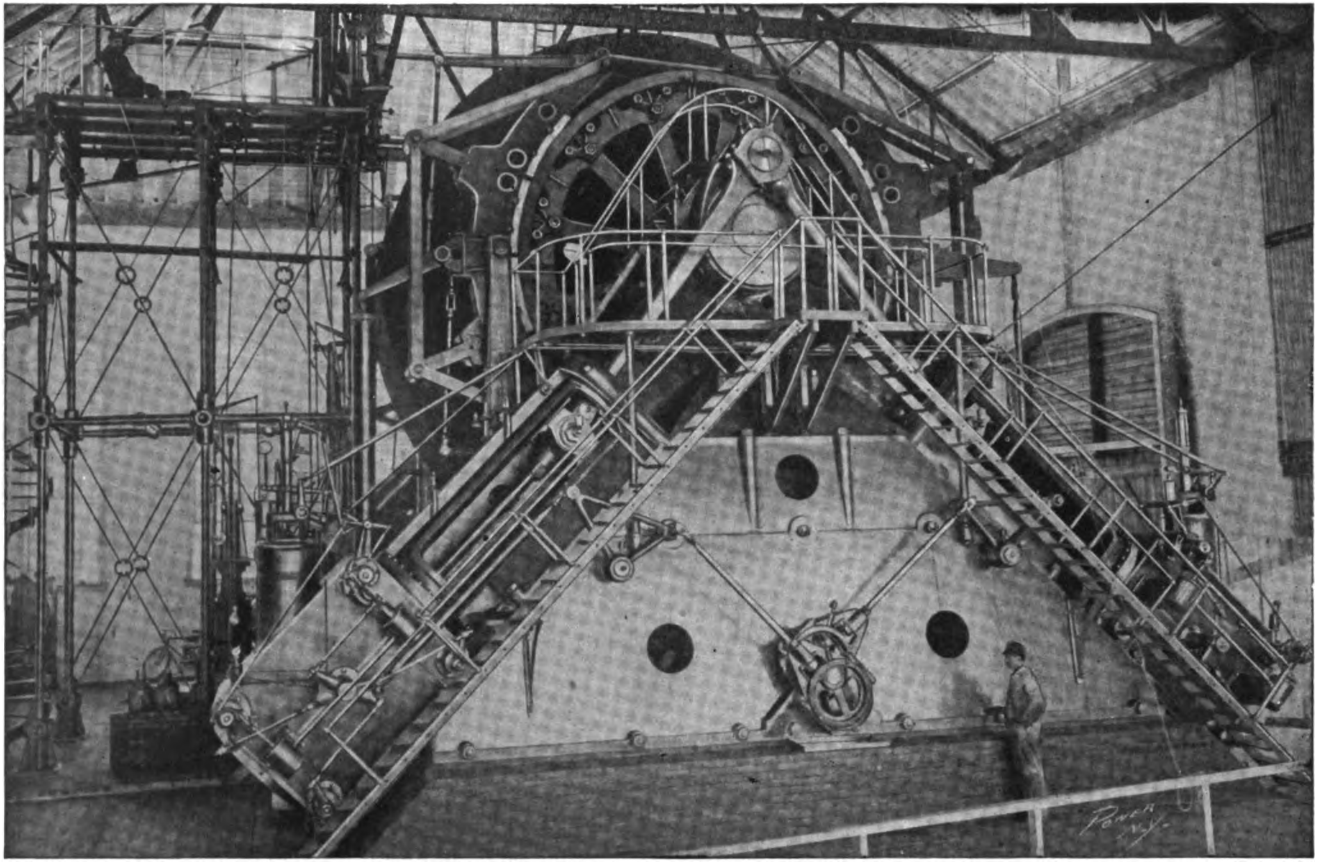


Fig. 4.

engine, release takes place before the end of the stroke, expansion is not complete, that is, the pressure at release is above that of the condenser, and the resistance of exhaust ports causes the back pressure to be above the actual condenser pressure. Moreover, the piston does not sweep the full length of the cylinder, and the clearance space must be filled with steam, which does little or no work. The theoretical and actual cards are shown in Fig. 4.

EFFICIENCY OF THE ACTUAL ENGINE.

We have seen that the efficiency of the theoretical engine is purely a thermal consideration; the efficiency of the actual engine,



TAMARAK HOISTING ENGINE AT MINE
Nordberg Manufacturing Co

however, is a mechanical matter. The measure of the activity of work is the horse-power which corresponds to the development of 33,000 foot-pounds per minute. As 778 foot-pounds are equivalent to one B. T. U., 33,000 foot-pounds, or one horse-power, is equivalent to $33,000 \div 778 = 42.42$ B. T. U. Now if a certain engine uses 84.84 B. T. U. per horse-power per minute, it is evident that its efficiency would only be $\frac{1}{2}$ or 50 per cent, because $42.42 \div 84.84 = \frac{1}{2}$. Hence we may say that the efficiency of the actual engine is equal to $\frac{42.42}{\text{B. T. U. per H. P. per minute}}$. This efficiency is always much less than that of the perfect engine. Let us now discuss the effects of some of the losses.

In the first place, the metal, being a good conductor of heat, becomes heated by the steam within and transmits this heat by conduction and radiation to the air or external bodies. With the cylinder well lagged much less heat is lost by radiation. If the lagging were perfect and the temperature of the cylinder remained the same as the temperature of the steam throughout the stroke, there would be no loss by radiation, but we should still lose heat by conduction to the different parts of the engine.

During expansion, the temperature and pressure of the steam decrease as the volume increases, and the temperature at exhaust is much less than the temperature at admission. In the perfect engine the working substance after exhaust is compressed to the temperature at admission, but in the actual engine much of this steam is lost and the compression of a part of it is incomplete, so that its temperature is less than the temperature at admission.

Suppose an engine is running with admission at 100 pounds absolute and exhaust at 18 pounds absolute. Then from steam tables we find the temperature at admission to be 327.6°, and at exhaust 222.4°. The metal walls of the cylinder, being good conductors and radiators of heat, are cooled by the low temperature of exhaust, so that the entering steam comes through ports and into a cylinder that is more than 100° cooler than the steam. This means that heat must flow from steam to metal until both are of the same temperature. This causes the steam to give up part of its latent heat, and as saturated steam cannot lose any of its heat without condensation, we find the cylinder walls covered with a

film of moisture known as initial condensation. This condensation in simple unjacketed engines working under fair conditions may easily be 25 per cent or more of the entering steam. The moisture in the cylinder has of course the same temperature as the steam; it has simply lost its heat of vaporization.

Although metal is a good conductor of heat it cannot give up nor absorb heat instantly; consequently during expansion the temperature of the steam falls more rapidly than that of the cylinder. This allows heat to flow from the cylinder walls to the moisture on them. As fast as the steam expands so that the pressure in the cylinder becomes less, this condensation will begin to evaporate. As the pressure falls it requires less and less heat to form steam, and therefore more and more of this moisture will be evaporated. At release the pressure drops suddenly, and more heat at once flows from the cylinder walls, and re-evaporation continues throughout the exhaust. Probably all of the water remaining in the cylinder at release is now re-evaporated, blows out into the air or the condenser, and is lost so far as useful work is concerned.

The steam that is first condensed in the cylinder does no work; its heat is used to warm up the cylinder, and later, when it is re-evaporated, it works only during a part of the expansion and at a reduced efficiency, because it is re-evaporated at a pressure and consequently at a temperature very much lower than that of admission. If the cut-off is short, perhaps 20 per cent of the steam condensed may be re-evaporated during expansion; if the cut-off is long, 10 per cent may be re-evaporated, the rest remaining in the cylinder at release still in the form of moisture. Thus some of the entering steam passes through the cylinder as moisture, until after cut-off, and still more passes clear through without doing any work at all.

Suppose an engine is using 30 pounds of steam per horse-power per hour and admission is at 100 pounds absolute. The latent heat of vaporization at this pressure is 884 B. T. U. per pound. If the condensation amounts to $33\frac{1}{3}$ per cent, then 10 pounds are condensed and we lose 10 times 884, equals 8,840 B. T. U. per hour, or 147.3 per minute; and since 42.42 B. T. U. represents 1 horse-power, we shall lose by condensation 147.3

divided by 42.42, equals $3\frac{1}{2}$ horse-power (nearly). If the cut-off is shortened, the condensation increases and may amount to 50 per cent at very short cut-off. Of course we use very much less steam at short cut-off than with long cut-off, and doubtless in many cases 50 per cent of the steam at short cut-off is not as great an absolute quantity as 30 per cent at long cut-off. Nevertheless, in all cases it is the percentages that go to make up the efficiency.

In addition to the actual loss from condensation in the cylinder there is still another loss due to the re-evaporation. Suppose, as before, that 10 pounds of steam are condensed in the cylinder, and that 20 per cent of this is re-evaporated during expansion. This will leave 8 pounds to be re-evaporated during exhaust. Suppose the exhaust is at 3 pounds above atmospheric pressure, or 18 pounds absolute (about). Then the heat of vaporization is 958.5 B. T. U. per pound of steam, and it will require 8 times 958.5, which equals 7668.0 B. T. U., to evaporate the 8 pounds. All of this heat is taken from the cylinder, leaving the engine much cooler than it would be were it not for this re-evaporation. This gives some idea of the great amount of heat passing away at exhaust, which is known as the exhaust waste.

In all cylinders it is necessary to have a little space between the cylinder cover and the piston when at the end of the stroke. In vertical engines the space is greater at the bottom than at the top. The volume of this space, together with the volume of the steam ports, is called the clearance. It varies from 1 to 20 per cent, depending upon the type and speed of the engine; the higher the speed, the greater the clearance. This clearance space must be filled with steam before the piston receives full pressure, but this steam does no work except in expanding, and the volume of the clearance offers additional surface for condensation.

Another important loss is that due to friction. We know that it takes considerable power to move an unloaded engine; if fitted with a plain, unbalanced slide valve, the power necessary to move the valve alone is considerable. The piston is made steam tight by packing rings, and leakage around the piston rod is prevented by stuffing boxes. All these devices cause friction as well as wear at the joints. The amount of power wasted in friction varies

greatly, depending upon the kind of valves, general workmanship, state of repair and lubrication.

MULTIPLE EXPANSION.

Two engines may be used together on the same shaft, partly expanding the steam in one of the cylinders, and then passing it over to the other to finish the expansion. One advantage from this arrangement is that the parts can be made lighter. The high-pressure cylinder can be of much less diameter than would be possible if the entire expansion were to take place in one cylinder. This, of course, makes the pressure exerted on the piston rod much less, and the piston rod and connecting rod can thus be made much lighter. The low-pressure cylinder must be larger than it otherwise would be, but its parts need not be much heavier, because the pressure per square inch is always low.

This arrangement gives not only the advantage of lighter parts, but a decided increase of economy over the single cylinder type. If attention is given to the matter, a loss of economy would be expected, because the steam is exposed to a much larger surface through which to lose heat, but the gain comes from another source, and is sufficient to entirely counterbalance the effect of a larger cylinder surface.

When very high pressure steam and a great ratio of expansion is used, the difference between the temperature of the entering and that of the exhaust steam is great. For instance, suppose steam at 160 pounds (gage) pressure enters the cylinder and the exhaust pressure is 2 pounds (gage), the difference in temperature is, from steam tables,

$$370.5^\circ - 218.5^\circ = 152^\circ.$$

This difference becomes nearly 230° if the steam is condensed to about three pounds absolute pressure. The cylinder and ports of the engine are cooled to the low temperature of the exhaust steam, and, as we have seen, a considerable quantity of the entering steam is condensed to give up heat enough to raise the temperature of the cylinder to that of the entering steam. As the ratio of expansion increases, the difference in temperature increases, and consequently the amount of steam thus condensed also increases. To keep this initial condensation

as small as possible we must limit the temperature, that is, we must not have as great a difference between admission and exhaust. To do this we must divide the expansion between two or more cylinders.

It will be remembered that the great trouble Watt found with Newcomen's engine was its great amount of condensation, and he stated as the law which all engines should try to approach, "that the cylinder should be kept as hot as the steam which enters it." This is to avoid condensation when steam first comes in. If, instead of expanding the steam in one cylinder, we expand it partly in one and then finish the expansion in another, we shall have passed it out of the first cylinder before its temperature falls a great deal, and consequently the cylinder walls will be hotter than they would be if we had expanded it entirely in one cylinder. This would then reduce the amount of steam condensed. The importance of this may not be evident at first, but it makes a great difference in the economy of the engine. If there is less condensation, there will be less moisture to re-evaporate, and consequently less exhaust waste; hence we shall save in two ways at the same time.

In a compound engine we admit the steam first to the smaller or high-pressure cylinder, and exhaust it to the larger or low-pressure.

Suppose steam at 160 pounds (gage) pressure is admitted to a cylinder, and the ratio of expansion is such that the steam is exhausted at about 60 pounds (gage) pressure; then the difference of temperature is

$$370.5° - 307° = 63.5°.$$

If, now, the steam when exhausted from the first cylinder enters a second and is allowed to complete its expansion, so that the exhaust pressure is about two pounds (gage) pressure, the difference of temperature in this cylinder will be

$$307° - 218.5° = 88.5°.$$

Then for the single engine, if the exhaust pressure is two pounds (gage), the difference of temperature is 152°, while in the compound engine this difference is divided into two parts,

63.5° and 88.5°. The cylinder condensation for both cylinders of the compound engine will be much less than if the total expansion took place in a single cylinder. The cylinders should be so proportioned that the same quantity of work may be done in each.

If there are two stages of expansion, the engine is called compound, three stages triple, and four quadruple.

Besides reducing the excessive condensation, there is still another gain in using multiple expansion. We have seen how much heat is lost by the exhaust waste, which in the simple

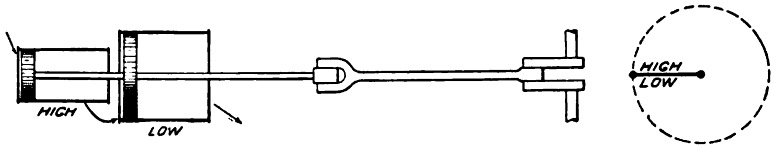


Fig. 5.

engine blows into the air or into the condenser, and is entirely lost. In the multiple expansion engine the exhaust and re-evaporation from one cylinder passes into the next and does work there; any

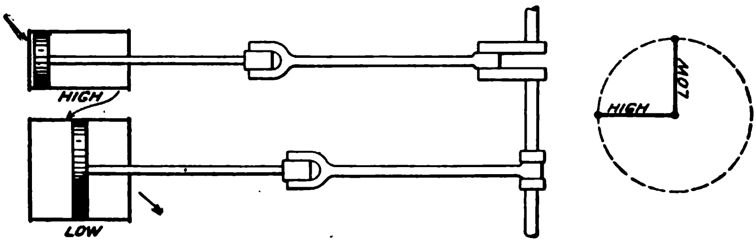


Fig. 6.

leakage from the high pressure is also saved, and does work in the low.

Mechanically there is a decided advantage; the several cranks give a more even turning moment, and the distribution of work between two or more cylinders makes it possible to use lighter individual parts; but there is a disadvantage in having more parts to look after, and a greater first cost of the engine.

Engines having two or more cylinders are arranged in various ways, but only the most common methods will be shown here. For two-cylinder compound engines the cylinders are often placed tandem, as shown in Fig. 5. This has the advantage of having but one crank, connecting rod and crosshead, but it has the disadvantage of dead points. In other words, the turning moment on the shaft is no more uniform than for a simple engine. When the cylinders are placed side by side and the cranks are at right

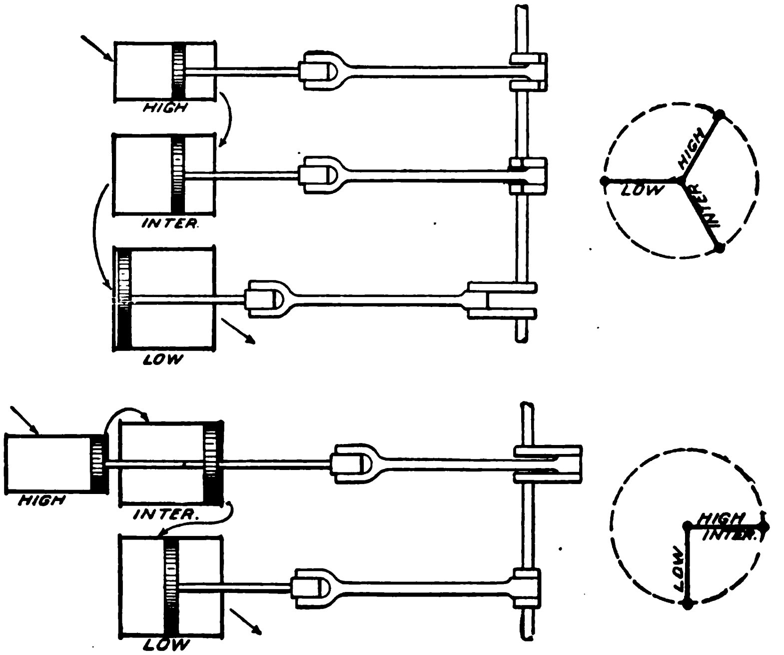


Fig. 7.

angles, as shown in Fig. 6, the low-pressure cylinder cannot receive the exhaust steam directly from the high-pressure cylinder; consequently a receiver must be used. In this case the advantage is due to the even turning moment, and the disadvantage is from the cost of receiver, extra crosshead, crank and connecting rod. In many instances, as electric lighting, marine work, etc., the advantage of the more nearly constant force acting on the shaft is worth far more than the extra first cost. The same may be said

of triple and quadruple expansion engines. (See Fig. 7.) Each type of engine has its special service, and what is suitable in one case may not be serviceable in another. Fig. 8 shows the diagram of a quadruple expansion engine, which is more commonly used in marine work.

JACKETING.

Another method of reducing the loss due to cylinder condensation is to supply heat to the steam while it is in the cylinder.

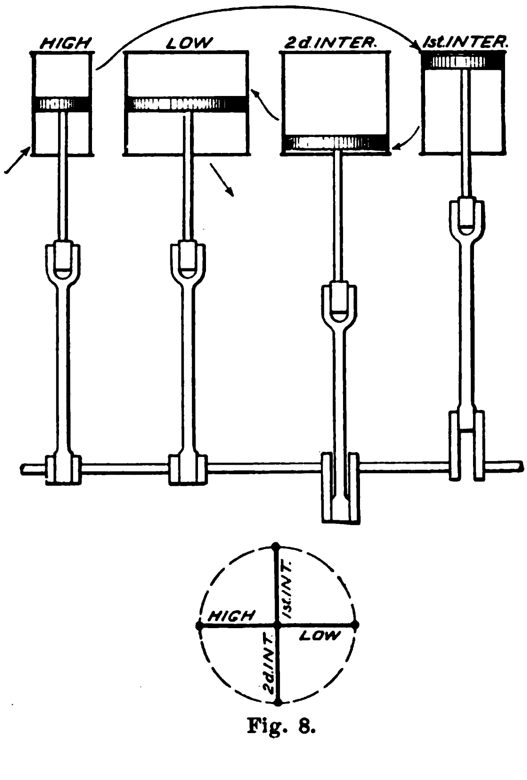


Fig. 8.

This is done by surrounding the cylinder with an iron casting and allowing live steam to circulate in the annular space thus formed. The cylinder covers are also made hollow to permit a circulation of live steam. A cylinder having the annular space (A, Fig. 9) filled with steam is said to be jacketed. A liner, L, is often used in jacketed cylinders.

The function of the jacket is to supply heat to the cylinder walls, to make up for that abstracted during expansion and exhaust, so that at admission the cylinder will be as hot as possible. The result is, that the difference in temperature between the cylinder walls and the entering steam is considerably less than in engines where no jacket is used. Condensation is therefore reduced, and since heat flows from the jacket to the cylinder during expansion, a much larger amount of this condensation is re-evaporated before release and it thus has a chance to do some work in the cylinder. This leaves a comparatively small amount of exhaust waste, and the heat thus abstracted is made good from the steam

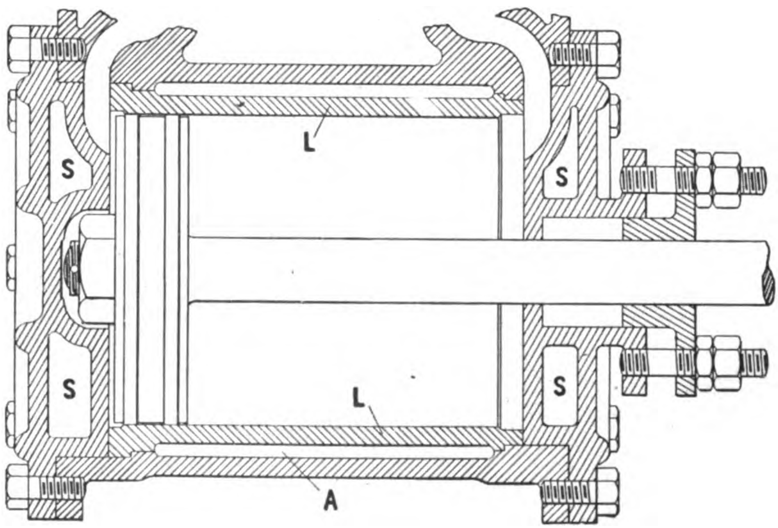


Fig. 9.

in the jacket. Since a large amount of heat is given up by the jacket steam a good deal of it must be condensed. Thus the question is asked: "What is the advantage of this method over that of allowing the entering steam to supply the heat by its own condensation?" This question is answered briefly as follows:

The loss of heat by condensing the steam would be less if the inside of the cylinder could be kept dry. We have seen how the moisture that collects by condensation is re-evaporated during expansion and exhaust, because the pressure falls and the cylinder walls are hotter than the steam. This re-evaporation takes place at the

expense of the heat in the cylinder walls, and they are thus cooled. We have already seen that a great many B. T. U.'s are thus taken from the cylinder and thrown out at exhaust at every stroke. Now if we can keep the inside dry, so that there will be little or no re-evaporation at exhaust, we shall make a considerable saving. The steam that condenses in the jacket does not re-evaporate; it is returned to the boiler as feed water, so that the only heat lost is the latent heat given up during condensation. If the cylinder is heated from within, both the latent heat given up by condensation and the latent heat required for re-evaporation are lost.

In a triple expansion engine there is one distinct advantage in allowing condensation in the cylinder, for this moisture acts as a lubricant, and as the heat of re-evaporation passes into the next cylinder and there does work, there is very little loss. In many marine engines there are no means of lubricating the cylinders at all, the engineers depending entirely upon the condensation.

SUPERHEATED STEAM.

We have already learned that superheated steam contains more heat than is necessary to keep it in the form of steam, and that consequently it can part with some of that heat without condensing.

Suppose this superheated steam is admitted to the cylinder of an engine. We know that the cylinder walls would be comparatively cool, as they were in contact with the exhaust steam and were still further cooled by the re-evaporation of water which was condensed on them at the previous admission of steam. Heat must flow from the steam to the walls until they are as hot as the steam. If the steam were merely saturated, as is usually the case, some of it would be condensed in order to supply this heat to the walls. But in the case of the superheated steam we know heat was given to it beyond that necessary simply to make saturated steam, so that it can give up some heat to the walls before it begins to condense.

Thus the amount of initial condensation is materially reduced, and consequently there will be less cooling of the cylinder walls by the re-evaporation of this condensation; therefore the walls will be hotter at the next admission, and it will require less heat to raise them to the temperature of the incoming steam.

Besides the saving in the engine there is a considerable gain due to the increased efficiency of the boiler, for by superheating, a part of the heat in the waste gases is utilized. If a boiler primes or is so constructed that it does not furnish dry steam, the gain from superheating will be most marked. If there is moisture in the steam and this passes over into the engine, the heat that it contains is entirely wasted, and the re-evaporation of this additional moisture during exhaust causes the cylinder walls to cool off still further.

Suppose we have steam in a boiler at 60 pounds pressure by the gage. Its temperature will be about 307° F. If this steam leaves the boiler by a pipe which passes through a furnace or chimney where the temperature is greater than 307°, it will become heated as it passes through. The pressure will, however, be the same, for it is still in communication with the boiler. This is actually the way in which steam is superheated in practical work.

One great difficulty in superheating is that the superheater, which is usually placed in the uptake, is subjected to a very high temperature and there is great danger of its burning out. The plates of a boiler are not likely to be damaged by the intense heat of the furnace, because the water by its good conducting power prevents their becoming overheated. But steam is a poor conductor of heat, and although the superheater is placed in the uptake or chimney, where the heat is not as intense, they nevertheless cause a great deal of trouble.

Superheating is efficient if low pressure is used, that is, steam at about 50 pounds pressure; but if saturated steam is at 150 to 200 pounds pressure, the temperature is high. If more heat is then added, the temperature becomes such that lubricants are decomposed. In such cases the piston and valve, not being lubricated, cut into the cylinder walls and the valve seat, thus causing leaks and requiring excessive power to move the engine. The packing in the stuffing boxes, unless metallic, is injured and causes considerable trouble. In case high-pressure steam is used the superheating must be slight, often so slight as not to be worth the trouble and expense.

CONDENSERS.

In our study of the theoretical engine on page 20, we learned that to meet the ideal conditions and to attain the maximum efficiency, steam must enter the engine at the constant temperature of the heat generator (or boiler) and must leave the engine at the constant temperature of the refrigerator (or condenser). We have also learned that the difference between the temperature of admission and exhaust is a measure of the thermal efficiency. In Part I of "The Steam Engine," Watt's principle was stated, namely, that "when steam was condensed it should be cooled to as low a temperature as possible." In our discussion of saturated vapors we also learned that the temperature varied with the pressure, and consequently to cool the condensed steam to as low a temperature as possible simply means to condense the steam to as low a pressure as possible.

In the ordinary noncondensing engine, steam cannot be expanded below a pressure of 14.7 pounds, because the atmosphere exerts that pressure at the opening of the exhaust pipe. In fact, this 14.7 pounds is only the theoretical limit, and in practice the exhaust is always a little above this because of resistance in the exhaust ports and exhaust pipe; 17 or even 18 pounds back pressure is more nearly the conditions of actual service.

During the forward stroke, steam expands from the pressure at admission to a much lower pressure at release; then the valve opens for the return stroke and on one side of the piston there is full steam pressure, and on the other side the pressure of exhaust, which acts against the piston and against the force of the incoming steam. If all of this back pressure could be removed so that there would be a vacuum on the exhaust side of the piston, the power of the engine would be increased by just so many pounds of M. E. P., and in addition to this the steam could expand to a very much lower pressure and therefore work with greater economy.

One pound of steam at 17 pounds absolute pressure occupies 23.22 cubic feet of space in the cylinder of the engine, but one pound of water in the condenser occupies only about 0.016 cubic foot, which makes the steam occupy nearly 1,450 times as much space as the water into which it condenses. If, then,

the exhaust steam could be condensed instantly, the back pressure would be reduced almost to zero and the engine would exhaust into a vacuum.

We know that a certain amount of heat is required to change one pound of water at a given temperature into steam at the same temperature; this is called the latent heat, or heat of vaporization. If the steam condenses, it must give up this latent heat. The easiest way of doing this is to let the steam mingle with a spray of water, as in the jet condenser, or come in contact with pipes through which cold water is circulated, as in the surface condenser. These two forms of condenser are fully described in Part I and will not be further considered here.

Unfortunately the mere condensation of the steam will not give a perfect vacuum, because more or less air, which is always in the water, comes over from the boiler and thus gets into the condenser. Moreover, the condensed water is hot, and a vapor rises from it in the condensing chamber; this, together with the air and some leakage, would spoil the vacuum were it not for the air pump, which removes the air and condensed steam. With the best air pump it would be impossible to maintain a perfect vacuum, but a vacuum of 26 inches, which corresponds to about 2 pounds absolute pressure, can readily be maintained in good practice.

Advantages of Condensing. It has already been stated that there is a gain in thermal efficiency by running an engine condensing, but it will be more clearly seen by considering a few figures. The thermal efficiency may be expressed by the formula:

$$E = \frac{T_1 - T_2}{T_1}.$$

This efficiency may be increased by making T_1 larger, which would happen if the boiler pressure were increased, or by making T_2 smaller, which would result from reducing the back pressure by condensing. If the boiler pressure is raised, both the numerator and denominator of the fraction will increase, and the value of the fraction will be but slightly greater. If, however, the back pressure is reduced, the numerator, $T_1 - T_2$, will be larger, while the denominator, T_1, will remain the same. It is apparent that this will cause a much greater increase in efficiency than raising the boiler pressure a like amount.

Suppose an engine is supplied with steam at 85.3 pounds (gage) pressure and it exhausts at 3.3 pounds (gage) pressure. The absolute temperature corresponding to $85.3 + 14.7 = 100$ pounds pressure is $327.58 + 461 = 788.58$, and the absolute temperature corresponding to $3.3 + 14.7 = 18$ pounds pressure is $222.40 + 461 = 683.40$. Then the thermal efficiency will be from the formula:

$$\frac{T_1 - T_2}{T_1} = \frac{788.58 - 683.40}{788.58} = .133 \text{ or } 13.3 \text{ per cent.}$$

If the boiler pressure were raised to 140 pounds absolute the efficiency would be

$$\frac{813.85 - 683.40}{813.85} = .16 \text{ or } 16 \text{ per cent.}$$

If instead of increasing the boiler pressure a condenser is used and thereby the exhaust pressure reduced to 4 pounds (absolute), the efficiency becomes

$$\frac{788.58 - 614.09}{788.58} = .221 = 22.1 \text{ per cent.}$$

Thus we see that if we lower the exhaust pressure 14 pounds we get a greater increase in efficiency than if the boiler pressure is raised 40 pounds.

Another method of showing the advantage by condensing the exhaust is by the indicator card.

Fig. 10 represents a card from a $12'' \times 20''$ engine making 75 revolutions with 75 pounds steam pressure. The dotted diagram represents a card taken when running without a condenser. Cut-off occurs at $\frac{1}{3}$ the stroke. The M. E. P. is 44.2 pounds. Hence the indicated horse-power is about 37.87.

The card shown in full line was taken with the same load, same speed, and with a condenser producing a vacuum of about 26 inches of mercury, which is equal to about 12.7 pounds. The absolute pressure at exhaust is then $14.7 - 12.7 = 2$ pounds. Since the load is the same, the areas and the M. E. P. must be the same in both cases. Cut-off is found to be only about $\frac{1}{5}$ the stroke. The above engine without the condenser uses an amount of steam represented by the length A C, while if a condenser is attached the steam used is represented by the length A C′. Thus

we see that the amount of steam consumed per stroke is considerably less if a condenser is used.

Quantity of Water. Besides merely condensing the steam the injection water cools it still further, so that more than merely the latent heat is removed from it. If exhaust steam enters the condenser at a temperature t_1, it contains a certain amount of heat, which is the *total heat* at that temperature. If it is condensed and cooled to a temperature t_2, at which it leaves the condenser, it then contains a certain amount of heat which is the *heat of the liquid* at this temperature t_2.

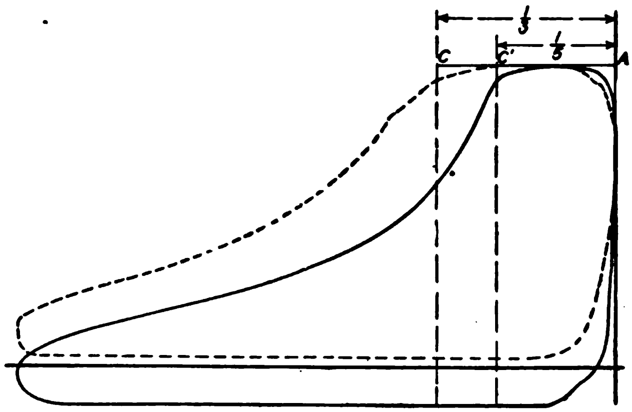


Fig. 10.

If A represents the total heat at t_1, and B represents the heat of the liquid at t_2, then the heat given up by one pound of condensed steam is equal to (A — B) B. T. U., provided the exhaust that enters the condenser is dry saturated steam. If C is the temperature of the injection water, and D is the temperature of the discharge water, then every pound of cooling water absorbs one B. T. U. for every degree rise in temperature; or we may say that the heat absorbed is equal to (D — C) B. T. U. per pound of cooling water. Then it will take as many pounds of water to absorb (A — B) heat units as (D — C) is contained times in (A — B). We may express this in terms of a formula thus:

$$W = \frac{A - B}{D - C}$$

In which W = pounds of cooling water per pound of steam.

For example: Suppose steam is expanded in an engine to 4 pounds absolute pressure. If the temperature of the injection water is 45°, and the condenser is of the surface type with discharge water at 120°, and the temperature of the condensed steam is 130°, how many pounds of injection water are required per pound of steam?

By consulting the steam tables we find the total heat of steam at 4 pounds pressure to be 1,128.6 B. T. U. The heat of the liquid in the condensed steam at 130° is 98.1 B. T. U. Then

$$W = \frac{1,128.6 - 98.1}{120 - 45} = 13.74 \text{ pounds.}$$

Suppose steam at 6 pounds absolute pressure exhausts into a jet condenser. The temperature of the injection water is 50° and the discharge is 120°. How many pounds of water are necessary to condense 8 pounds of steam?

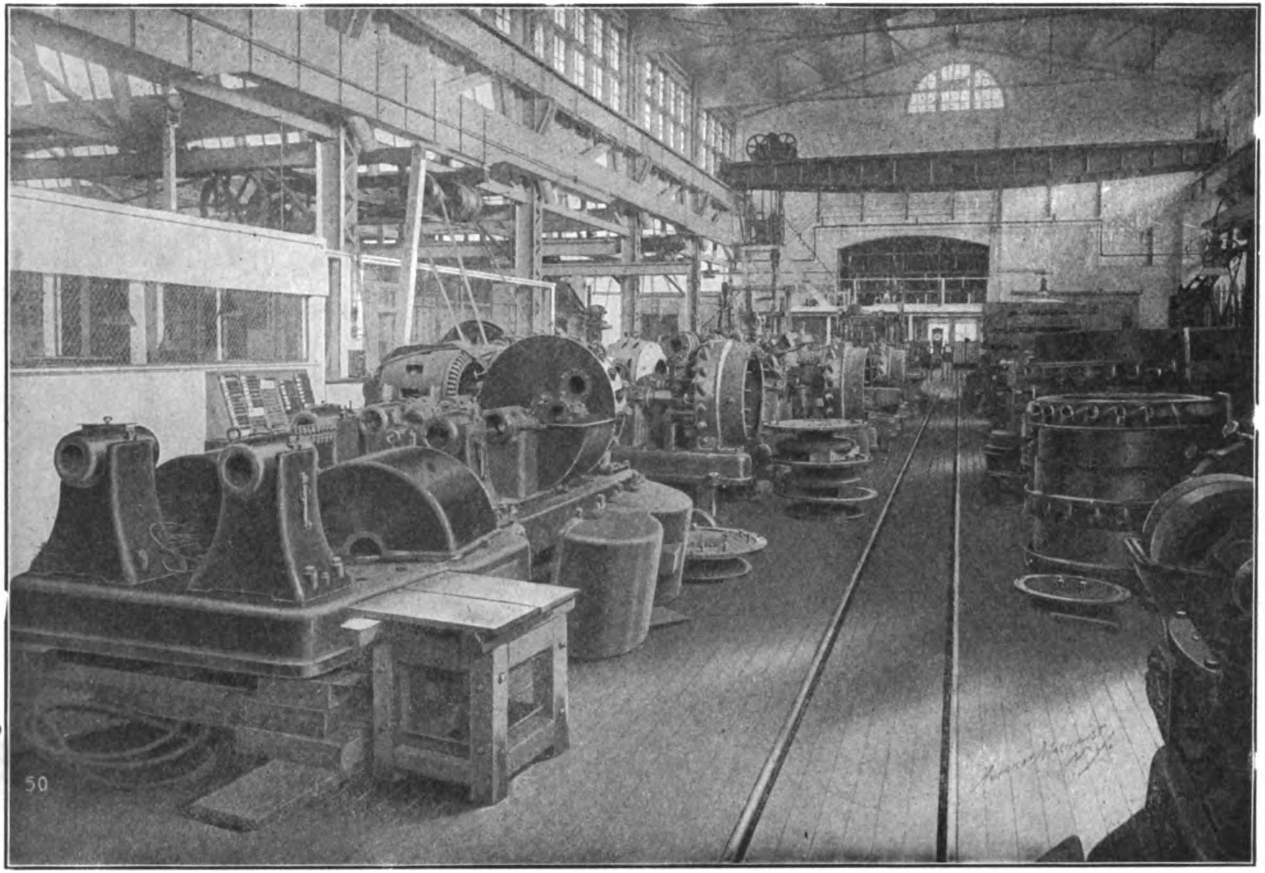
In the jet condenser the temperature of the condensed steam and discharge water is the same. We find from the steam tables that the total heat of steam at 6 pounds absolute is 1,133.8 B. T. U., and the heat of the liquid in the condensed steam at 120° is 88.1 B. T. U. Then, as before,

$$W = \frac{1,133.8 - 88.1}{120 - 50} = 14.94.$$

For 8 pounds it will take $14.94 \times 8 = 119.52$ pounds.

The above calculation cannot be relied upon to any great extent, for we seldom know the true conditions in the condenser, and it would be of little value to us if we did know, as the exact conditions will change considerably. In practice it is customary to allow for about twice as much water as the above calculation would require. These figures give us a fair idea of the necessary sizes of pipes and passages leading to the condenser, and give a basis for estimating the dimensions of the air pump.

Cooling Surface. The amount of surface required to condense the steam in surface condensers depends upon the efficiency of the metal, the condition of the tubes, the difference in temperature between the two sides and their thickness. There have been no satisfactory tests to determine the amount of cooling surface



ERECTING SHOP FOR LARGE STEAM TURBINES.
De Laval Steam Turbine Co.

necessary for a condenser, and in actual practice there seems to be a wide diversity of opinion. The tubes of a condenser are much thinner than boiler tubes, and much more clean, hence we might expect them to be more efficient in condensing the steam than the boiler tubes are in evaporating it. Boilers may evaporate 7 pounds or more of steam per hour, per square foot of heating surface. Seaton, an eminent authority on marine work, says that with cooling water at 60° F, and the discharge at 120°, a condensation of 13 pounds of steam per square foot per hour is fair average work. A new condenser will of course condense much more than this. If the exhaust pressure is from 6 pounds to 10 pounds absolute, an allowance of 1.5 to 1.8 square feet of cooling surface may be allowed per indicated horse-power, depending upon the pressure. This assumes that the temperatures of the injection and discharge water shall be 60° and 120° respectively.

It is evident that the amount of surface will depend upon the quantity of steam used per hour by the engine, the pressure and temperature of the exhaust and the temperature of the cooling water and discharge. There must also be an allowance for inefficient work after the condenser has become fouled with service. All these conditions make the problem so uncertain that calculations by means of formulæ are likely to be untrustworthy, and it is best at all times to make estimates from the figures given for similar conditions in actual service.

Measurement of Vacuum. We have seen that in order to maintain a vacuum in the condenser it is necessary to pump out, by means of an air pump, the air that leaks in. Evidently, if we are to maintain a proper vacuum, it is necessary to know at all times just how much pressure there is in the condenser. If the pressure increases, the air pump can be run a little faster until the proper vacuum is obtained.

From the study of pneumatics we know that the pressure of the atmosphere can be measured by means of a column of mercury. The atmospheric pressure will support a column of mercury about 30 inches high, which is equivalent to a pressure of 14.7 pounds (nearly). It would be inconvenient to attach a mercury column to the condenser and so we use a gage, in general appearance similar to a boiler gage except that the dial is graduated to read

in inches of mercury instead of in pounds pressure, and it indicates the inches of vacuum *below atmospheric pressure.* If the pointer of the vacuum gage stands at 20, it means that the pressure is equal to 20 inches below atmospheric pressure. Since 30 inches is equal to 14.7 pounds, 20 inches would be equal to $\frac{20}{30} \times 14.7 = 9.8$ pounds. This is 9.8 pounds below atmosphere, or $14.7 - 9.8 = 4.9$ pounds above zero.

If the vacuum gage stands at 26 inches, what is the absolute pressure in the condenser?

$$\frac{26}{30} \times 14.7 = 12.74 \text{ pounds below atmosphere}$$

$$14.7 - 12.74 = 1.96 \text{ pounds absolute pressure.}$$

EXAMPLES FOR PRACTICE.

1. Steam enters the condenser at 11 pounds pressure (absolute). The water enters at 53° and leaves at 110°. The condensed steam leaves at 120°. If the engine uses 26 pounds of steam per horse-power per hour while running at 53 horse-power, what would be the theoretical amount of water used by the condenser and what should be the area of cooling surface of the condenser?

Ans. $\begin{cases} \text{106 square feet.} \\ \text{25,479 pounds} \end{cases}$

2. Steam enters a jet condenser at 16 pounds absolute pressure. The injection water has a temperature of 48°. If the temperature of the discharge water is 115°, how many pounds of injection water are necessary per pound of steam?

Ans. 15.89 pounds.

3. If the pointer of a vacuum gage stands at 9, what is the approximate absolute pressure in the condenser?

Ans. 10.3 pounds.

4. The pointer of a vacuum gage which is attached to a condenser stands at 23. What is the approximate pressure in the condenser?

5. The steam gage indicates 175 pounds and the vacuum gage 26 inches of mercury. What is the total difference in pressure?

Ans. 187.74 pounds.

OTHER DEVICES.

Corliss Valves. From a thermal point of view the advantages of the Corliss valve may be summed up in a few words. The exhaust valves are separate from the admission valves and hence the exhaust steam does not come in contact with the admission ports. Thus the admission ports are not cooled and there is less condensation in proportion to the ratio of expansion than in the plain slide valve type of engine. The short ports reduce the volume of clearance and thus save clearance steam and reduce the surface exposed to condensation.

Separator. We have studied the loss of heat due to re-evaporation at exhaust and find that it is considerable. The more moisture there is in the entering steam, the more moisture there will be to re-evaporate; consequently more heat will be lost from the cylinder. It will at once be seen that if we can separate the moisture from the entering steam, we shall keep considerable water out of the cylinder.

PISTON SPEED.

The total distance passed over by the piston in one minute is called the piston speed. As the piston travels the length of the stroke twice for every revolution, the piston speed is evidently equal to the length of stroke multiplied by twice the number of revolutions. Then in the formula for horse-power, $\frac{PLAN}{33,000}$,

$$N = \text{the number of strokes}$$
$$L \times N = \text{the piston speed.}$$

In case the stroke is stated in inches, the piston speed in feet equals $\frac{L \times N}{12}$.

In determining the allowable piston speed, local conveniences, durability and the character of the work to be done should be considered. Other things being equal, the piston speed increases slightly with an increase in the length of stroke.

The piston speeds in common practice are as follows:

Direct acting pumping engines	100 to 150 feet per minute
Small stationary engines	300 to 500 feet per minute
Large stationary engines	500 to 900 feet per minute
Corliss engines	400 to 800 feet per minute
Locomotives	700 to 1,200 feet per minute
Marine engines	750 to 1,000 feet per minute

RATIO OF EXPANSION.

The ratio of expansion as usually understood is the piston displacement divided by the volume of the cylinder to the point of cut-off. Thus if the piston is 10 inches in diameter and the stroke is 12 inches, the piston displacement is $10^2 \times .7854 \times 12 = 942.48$ cubic inches. If cut-off occurs at $\frac{1}{3}$ the stroke, or when the piston has moved 4 inches, the volume to cut-off is $10^2 \times .7854 \times 4 = 314.16$ cubic inches, and the ratio of expansion is $\frac{942.48}{314.16} = 3$. This may be stated thus:

$$\frac{\text{area of cylinder} \times \text{length of stroke}}{\text{area of cylinder} \times \text{distance to cut-off}} = \text{ratio of expansion.}$$

Since the area of the cylinder appears in both numerator and denominator, we may cancel and write:

$$\text{Ratio of expansion} = \frac{\text{length of stroke}}{\text{distance to cut-off}}.$$

If clearance is taken into account, the true ratio of expansion is different from the apparent ratio as found above. The steam which expands in the cylinder is not merely the volume equal to the piston displacement, but is this volume plus the volume of the clearance.

With a late cut-off and small clearance the true ratio of expansion differs but little from the apparent. But when the cut-off is early and the clearance is large there is considerable difference, as shown in Fig. 11.

In this diagram D E represents the stroke, and is $2\frac{1}{2}$ inches long. D′ D represents the clearance volume, which is equal to $\frac{1}{8}$ the piston displacement, and is therefore .31 inch long. Let cut off occur at $\frac{1}{8}$ of the stroke; then the apparent ratio of expansion is equal to $\frac{D\,E}{A\,C} = \frac{2\frac{1}{2}}{.31} = 8$. If we consider the clearance, the diagram is changed, as shown by the dotted lines, and the true ratio of expansion is

$$\frac{D'\,E}{A'\,C} = \frac{2.80}{.62} = 4.5.$$

Thus we see that the ratio of expansion is not 8 but 4.5.

WORK DONE IN THE CYLINDER.

The amount of work done per stroke in the engine cylinder is usually found by means of an indicator. It is often desirable, especially in designing engines, to know the work that may be expected under given conditions before the engine is built. The work done in the cylinder is proportional to the mean effective pressure, so that if we are to determine the probable power of our engine we must in some way ascertain the probable mean effective pressure. If the initial pressure, the back pressure and the ratio of expansion are known, we can find the probable M. E. P. by

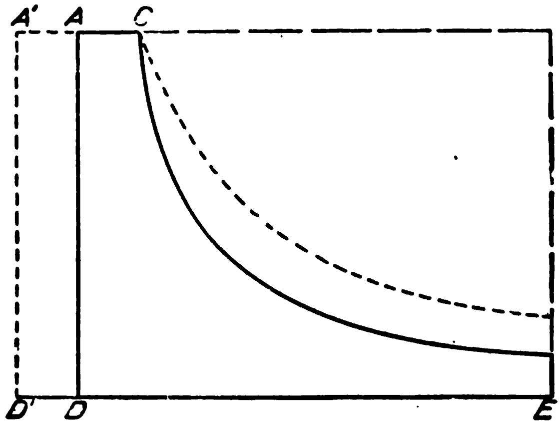


Fig. 11.

means of a formula. This formula does not provide for the losses due to compression, clearance, condensation and leakage, so that the value thus obtained must be multiplied by a factor depending upon the type of engine.

Let P = absolute initial pressure
R = ratio of expansion
p = back pressure (absolute)

Then neglecting all losses, the theoretical M. E. P. will be expressed by the formula,

$$\text{M. E. P.} = \text{P}\,\frac{(1 + \text{hyp. log. R})}{\text{R}} - p.$$

In order to allow for the losses due to compression, clearance

condensation, wire-drawing, etc., it is customary to multiply this value by factors given in the following table:

Engines with special valve gear and independent cut-off valves, about	.90
Plain slide-valve engines	.75 to .90
Compound engines	.65 to .80
Triple expansion engines	.50 to .70

These factors are for unjacketed engines; with jackets the factor might be .05 more in each case. On account of wire-drawing and large compression, high-speed engines would have the smaller value in its respective class. Only the best-designed engines with good valve gears and under favorable conditions will warrant the use of the larger factor in their respective classes. Large engines will of course have larger factors than small engines.

Example.— What is the probable M. E. P. when the initial pressure is 95.3 pounds (gage), the back pressure 3 pounds (absolute) and the ratio of expansion 4? The engine is large, of the plain slide-valve type with steam jackets.

$$P = 95.3 + 14.7 = 110 \text{ pounds}$$
$$p = 3 \text{ pounds}$$
$$R = 4$$

From the following table, hyp. log. R = 1.3863

$$\text{M. E. P.} = \frac{110\,(1 + 1.3863)}{4} - 3$$

$$= 62.62 \text{ pounds.}$$

For a large plain slide-valve engine, unjacketed, we might expect a factor of .85 to .90, depending upon the conditions. Suppose we call the factor .87, then adding .05 for the effect of the steam jacket we get .92.

$$62.62 \times .92 = 57.6 \text{ pounds as the probable M. E. P.}$$

A rough approximation to the power to be expected of a compound, or triple expansion, engine, may be found in a similar manner by assuming that all the expansion takes place in the low-pressure cylinder, neglecting losses as before.

Example.— Find the probable I. H. P. for a triple expansion engine with the following dimensions and data:

Revolutions,	115 per minute
Initial pressure,	140.3 pounds (gage)
Back pressure,	2 pounds (absolute)
Total ratio of expansion,	12
Diameter of low-pressure cylinder,	60 inches
Stroke,	30 inches

$$\text{M. E. P.} = \frac{P\,(1 + \text{hyp. log. R})}{R} - p$$
$$= 155\,\frac{(1 + 2.48490)}{12} - 2$$
$$= 43.01.$$

In practice only about .65 of this is actually obtained, so the probable M. E. P. is $.65 \times 43.01 = 27.95$ pounds.

The I. H. P. is now found from the expression $\frac{PLAN}{33{,}000}$ as follows:

$$\text{I. H. P.} = \frac{27.95 \times 2\frac{1}{2} \times 2{,}827.4 \times 2 \times 115}{33{,}000} = 1{,}377.$$

TABLE OF HYPERBOLIC OR NAPIERIAN LOGARITHMS.

No.	Log.	No.	Log.	No.	Log.	No.	Log.
1.25	.22314	3.75	1.32175	6.25	1.83258	8.75	2.16905
1.50	.40546	4.00	1.38629	6.50	1.87180	9.00	2.19722
1.75	.55962	4.25	1.44692	6.75	1.90954	9.25	2.22462
2.00	.69315	4.50	1.50407	7.00	1.94591	9.50	2.25129
2.25	.81093	4.75	1.55814	7.25	1.98100	9.75	2.27726
2.50	.91629	5.00	1.60943	7.50	2.01490	10.00	2.30258
2.75	1.01160	5.25	1.65822	7.75	2.04769	12.00	2.48490
3.00	1.09861	5.50	1.70474	8.00	2.07944	15.00	2.70805
3.25	1.17865	5.75	1.74919	8.25	2.11021	18.00	2.89038
3.50	1.25276	6.00	1.79176	8.50	2.14006	20.00	2.99570

CRANK ACTION.

In the steam engine the steam exerts a pressure on the crank pin through the piston rod and connecting rod. When the crank is at the dead center, the entire pressure is on the bearing of the crank shaft; there is no tendency to turn the crank. In any other position the steam pressure tends to turn the crank pin. As the crank pin moves from dead center, the tendency increases until it

reaches a maximum and then decreases until, at the other dead center, it is zero again. If the connecting rod were of infinite length, and steam were admitted throughout the whole stroke, the maximum tendency, or the maximum turning moment as it is called, would occur with the crank at right angles to the line connecting the dead points.

In the actual engine the thrust along the rod is constantly varying even though the pressure on the piston remains the same. This is due to the angularity of the connecting rod. The turning moment is always equal to the thrust along the connecting rod multiplied by the perpendicular distance from the connecting rod to the center of the shaft. If the steam pressure on the piston remains constant, the maximum turning moment occurs

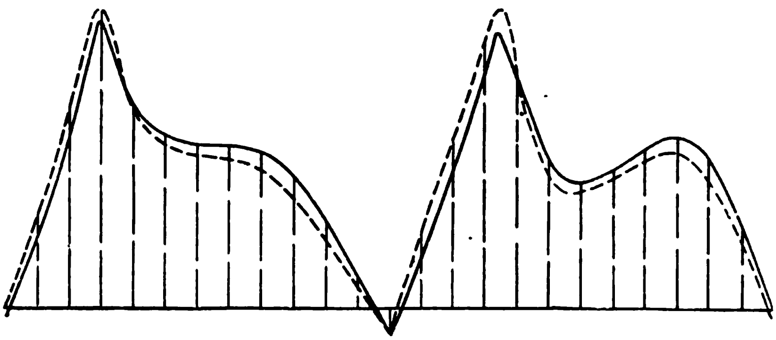


Fig. 12.

when the connecting rod is at right angles to the crank, for in this position the perpendicular distance from rod to center of shaft is a maximum equal to the length of the crank, and as the rod makes its greatest angle with the line connecting the dead center at this point the thrust along it will also be a maximum. If the cut-off is very early, $\frac{1}{4}$ stroke for instance, the maximum thrust along the rod will occur earlier than at the point previously mentioned, but the leverage of the force will be less, so that really there will be little change in the point of maximum turning moment no matter where the cut-off may occur.

To represent this turning moment, diagrams of crank effort are drawn. These diagrams may be drawn with rectangular co-ordinates having the crank angles represented as abscissae and

the turning moments corresponding to these angles as ordinates.

Besides the thrust of the connecting rod we must take into account friction and the inertia of the reciprocating parts. At first this may be thought of small consequence, but with a fairly heavy piston and connecting rod we can easily see that at high speed the momentum would be great. On a vertical engine, on the up stroke the steam has to lift this heavy mass and impart a very considerable velocity to it, while on the down stroke the acceleration of the mass is added to the steam pressure. This makes the effective force on the up stroke less than that due to the actual steam pressure, and greater on the down stroke. The crank effort diagram represented in Fig. 12 is from a horizontal engine of practical proportions. The initial steam pressure is 50 pounds per square inch. Cut-off at $\frac{1}{6}$ stroke. The engine makes

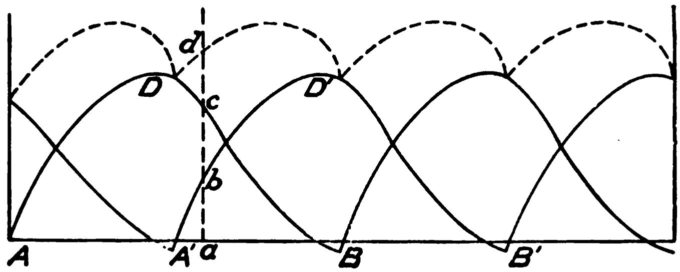


Fig. 13.

240 revolutions per minute. The dotted lines represent the crank effort, without considering friction. The full line is the diagram when weight of moving parts and inertia are considered.

In drawing a crank effort diagram, friction is often neglected, but the effect of inertia of the moving parts is of great importance, especially in the case of high speed.

It has been mentioned in connection with compounding that if there are two or more cranks on the shaft the turning moment is more nearly constant. We can now see that this is so. In Fig. 13, A D B represents the curve of turning moments on one crank, and A′D′B′ the curve of turning moments on the other crank, which is at right angles to the first. To find the total turning moment on the shaft the dotted line curve is drawn.

The points are obtained by adding the ordinates. Thus, to find the point *d* add *ac* to *ab*. Then $ad = ac + ab$. It is easily seen that the dotted line curve is more nearly a straight line than the curve A D B.

ECONOMY OF THE STEAM ENGINE.

The effects of various devices and methods employed to increase the economy of the steam engine cannot be studied from the theoretical side alone if we wish to obtain satisfactory results So complicated and important is the action of the cylinder walls, that if we would learn the conditions favorable to economy we must study the actual tests of engines in service conditions.

Effect of Raising Steam Pressure. In general there is a gain in thermal efficiency by increasing the steam pressure and the total number of expansions, provided proper means are taken to lessen the undesirable effects due to increased condensation and re-evaporation caused by the increased expansion. The initial condensation, of course, places a pretty strict limit upon the number of expansions profitably used in a simple expansion engine.

If we raise the steam pressure and keep the same cut-off, the conditions will be slightly different. The losses from external radiation increase with the rise of steam pressure, but the horsepower increases more rapidly, hence there is a net gain, and since the changes of temperature are greater, the higher the pressure, it is reasonable to suppose that the loss from condensation and re-evaporation will increase. It is not difficult to imagine a point where these losses may offset the gain, and where, indeed, if the pressure is raised too far there will be a net thermal loss. This has actually happened in some cases. At first the gain from raising the steam pressure is rapid. As the pressure rises, the gain increases more slowly, and finally it is not worth the expense, if, indeed, there is not actual loss.

If we bear in mind that there is little gain in economy to be obtained by increasing the steam pressure beyond a certain moderate limit, unless the ratio of expansion is also increased, and that the losses from condensation, re-evaporation and exhaust waste limit the number of expansions profitably used in a simple expansion engine, we shall at once see that we must look to other devices

for decided gains in economy. The most important of these have already been briefly considered.

Superheated Steam. Tests show a very decided gain in economy from using superheated steam, yet practically no permanent results have been obtained, on account of the difficulty of maintaining the superheating apparatus. To superheat to the best advantage we must have a coil of pipe in the flue and thus make use of the otherwise wasted heat. But the intense heat soon burns out the metal and together with the great pressure it is rapidly wasted away. Gains of from 15 per cent to 20 per cent from using superheated steam have actually been observed on simple engines. There has also been some gain noted by the use of superheated steam in triple expansion engines, but the gain is less than in the simple type, because naturally there is less condensation in the triple than in the simple engine.

Steam Jackets supply a small amount of heat to the cylinder during expansion which can be converted into work, but the chief advantage of a jacket is that it keeps the inner walls of the cylinder warmer at admission and thus reduces initial condensation and saves much loss of re-evaporation at exhaust. It will be evident that a large part of the heat of the steam jacket flows to the cylinder during exhaust and is thus entirely lost in the simple engine. In the triple engine this heat passes into the intermediate and low-pressure cylinders; consequently we might expect a greater gain from using a jacket on a triple than on a large simple engine. The main advantage of the jacket has been previously pointed out, and it may be stated that in all cases the gain from the jacket is small and there is to be found a considerable diversity of opinion as to its real advantages. On some engines there is undoubtedly little if any gain. The largest gain is in the smaller engines of say under 200 H. P.; on very small engines the gain is quite large, having been as much as 30 per cent on a 5″ × 10″ engine when developing only $1\frac{1}{2}$ H. P. under light load. On a 10 H. P. engine the gain might be as much as 25 per cent. On engines of about 200 H. P. the gain would probably be 5 per cent to 10 per cent for simple condensing and compound condensing, and from 10 per cent to 15 per cent for triple expansion. The saving on large engines, of say 1,000 H. P., is very small, the reason being

that large engines offer less cylinder surface per unit of volume than small ones, and hence we find proportionately less cylinder condensation in large engines than in small ones. The very small engines, in which the gain would be greatest, are seldom jacketed, because they are built for inexpensive machines and the first cost is of more consequence than the later economy.

Compounding is the most effective method of increasing the number of expansions and at the same time avoiding excessive cylinder condensation. We know that increase in boiler pressure and increase in expansion in simple engines are economical only to a certain limit. We shall now discuss the gain due to compounding and the conditions under which it is advantageous.

A direct comparison between tests of different engines is impossible because of the different steam pressures, etc., but a careful study seems to show that for simple condensing engines there is no advantage in raising the steam pressure above 80 pounds. In compounding, the pressure can be advantageously raised to 135 pounds. The gain due to the higher pressure, greater number of expansions and compounding may be 20 per cent to 30 per cent. For triple-expansion engines the most economical steam pressure is of course higher than for compound and may be used to advantage up to 180 pounds or more. The gain from using a triple over a compound may be about 5 to 10 per cent or more. These figures are good only for engines under full load and proper point of cut-off. A compound will usually suffer more loss of economy under light load than a simple, and the triple will suffer more than a compound.

Cut-off and Expansion. The best point for cut-off for a simple engine, whether jacketed or not, is about $\frac{1}{3}$ stroke if the engine is noncondensing and about $\frac{1}{6}$ stroke if condensing. The total expansions of a triple engine or a compound is commonly known as

$$\frac{\text{ratio of low pressure to high pressure}}{\text{fractional part of stroke at cut-off in high}}.$$

For instance, if the cylinders are as 1 : 3 : 8, and cut-off is $\frac{2}{5}$ stroke in high, then the total expansions as conventionally used is $\frac{8}{\frac{2}{5}} = 20$. For the best service of triple engines on land 20

expansions are used. The conditions of service make it impossible to use as many expansions in marine work, hence the relatively poor economy of marine engines. For compound engines 15 expansions seems to be the best in general. Of course it must be understood that the type of engine and conditions of service may necessitate different arrangements, but when such is the case a less economy of steam consumption usually results.

Variation of Load. An engine should be designed to give a fair economy over a reasonable variation of load. Ordinarily if it gives its best economy at normal load there will be a sufficient range. Usually, it happens that the best mechanical efficiency is obtained with a little longer cut-off than that which shows the most economical steam consumption per indicated horse-power. This is because, as the cut-off lengthens, the power increases faster than the frictional losses. It will be evident from this that increasing the cut-off slightly reduces the thermal efficiency, but the gain in mechanical efficiency may offset the thermal loss. Shortening the cut-off causes loss of both thermal and mechanical efficiency.. It would then seem that if the engine were given a little longer cut-off than that which would produce the best thermal efficiency at normal load, the power could be reduced with less actual loss. For a slight reduction we should lose mechanical efficiency and gain thermal, and for a slight increase we should gain mechanical and lose thermal. Thus there would be a wider range with good results. There is always more loss by decreasing cut-off below the point of maximum efficiency than by lengthening it. That is, the engine works at a greater disadvantage when running under a light load than when running under a heavy one.

The allowable range of load for a simple engine is greater than for a compound or triple. If the power of a compound is reduced by shortening the cut-off of the high-pressure cylinder without shortening the low proportionally, there is likely to be an uneven distribution of work and, consequently, a wide fluctuation of temperature, which will cause so much condensation as to offset the advantages of compounding. Moreover, the large expansion in the high-pressure cylinder may reduce the admission pressure to the low cylinder so much that the expansion in the low may be carried below the atmosphere in a noncondensing engine and thus

cause a loop at the end of the card, as we have learned in "Steam Engine Indicators." This loop means a loss of power, and that the high-pressure piston is dragging the low and the engine is using up power on itself as it were. A triple engine is even more troublesome than a compound, and besides giving the trouble already mentioned, the dragging of the low-pressure piston may injure the cylinder and loosen the guides. There is little difficulty in increasing the power of a compound or triple. A simple engine may easily be run at reduced power, either by shortening the cut-off or reducing the steam pressure or both.

Effect of Speed. The transfer of heat between the steam and walls of the cylinder, although very rapid, is not instantaneous. The longer the steam can remain in contact with the cylinder walls, the more heat will be lost. Hence it is reasonable to suppose that an increase of speed will reduce condensation. This has been found by tests actually to be the case. Other things being equal, a reasonable increase of speed results in better economy, but as we can seldom get as good a valve motion at high speed, we may lose almost as much in this way as we gain by decreasing our condensation. The most efficient valve gears are to be found on slow-speed engines, and hence we usually find that the most economical engines are slow speed. Nevertheless, with a given valve gear, an increase of speed gives better economy up to the limit of good valve motion. High-speed engines also require more clearance, which means another loss. These two factors, valve motion and clearance, limit our increase of speed beyond a certain point, just as condensation will limit our indefinite increase of boiler pressure.

Feed Water Heaters. The use of feed water heaters has been discussed in the instruction papers on boilers, but it may be well to observe here that in many places where water is expensive condensing engines cannot be run; nevertheless, a very considerable saving can be effected by allowing the exhaust steam to condense in a feed water heater, and thus save the heat that would otherwise be wasted, or the exhaust steam may be used for heating purposes. Of course in such cases the steam consumption of the engine is high, but if proper allowance is made for the heat used for other purposes, the actual fuel consumption rightfully charged to the engine is not excessive. If the feed water is heated by

waste gases, then the gain belongs to the boiler and not to the engine.

The following represents the average steam consumption in pounds per I. H. P. per hour for various types of engine:

Triple expansion stationary	11 to 14 pounds
Triple expansion marine	13 to 16 pounds
Compound stationary	12 to 15 pounds
Compound marine	18 to 21 pounds
Simple condensing	17 to 20 with jacket
Simple condensing	18½ to 22 without jacket
Simple noncondensing	24 to 33 or more

All these figures have been taken from the results of careful tests of engines in actual service, and represent good engines of their respective types.

Direct-acting steam pumps, although very important engines, are extravagant in their consumption of steam. The absence of a fly wheel makes it almost impossible to use steam expansively, and hence the pump takes steam full stroke.

The steam consumption for a simple direct-acting steam pump may be anywhere from 60 to 200 pounds per horse-power per hour, depending upon the size and conditions of service.

The economy of an engine is usually stated in terms of steam per horse-power per hour, or of B. T. U. per horse-power per minute. The latter is more useful for purposes of comparison with other engine tests, but the real criterion of economy is the actual cost of power in terms of fuel consumption, for this can at once be reduced to dollars and cents, and the manufacturer or the builder of the engine can determine the effect of devices for economy in terms of money, which to him is worth more than B. T. U. It may often happen that some expensive device, while effecting a saving of heat consumption, may not save fuel enough to pay for its first cost, owing to the peculiar conditions of service.

TESTING.

The principal object in testing a steam engine is to determine the cost of power or the effect of such conditions as superheating, jacketing, etc., upon the economy of the engine. We must therefore, in the first case, measure the cost of fuel, and in

the second the actual heat used. In either case we must calculate the power of the engine.

The indicated power is determined in the familiar way by means of the indicator, and the actual power by means of a dynamometer or friction brake. (For further particulars of these instruments and apparatus see "Steam Engine Indicators.") To determine the cost of power in terms of coal it is necessary to conduct a careful boiler test, usually of twenty-four hours duration.

When the cost is expressed in terms of steam per horse-power per hour, we may follow either of two methods. We may condense and weigh the exhaust steam, or we may weigh the feed water supplied to the boiler. An hour under favorable conditions is usually sufficient for such tests. Steam used for any purpose other than running the engine must be determined separately and allowed for.

Probably the most accurate terms in which to state the performance of an engine is in B. T. U. per horse-power per minute. When the cost is expressed thus, it is necessary to measure the steam pressure, amount of moisture in the steam, and temperature of condensed steam when it leaves the condenser. Jacket steam must be accounted for separately. Important engines with their boilers, etc., are usually built under contract to give a certain efficiency, and their fulfilment of this contract can be determined only by a complete test of the entire plant. Before beginning the test, the engine should be run for a short time in order to limber it up and get it thoroughly warmed. It is of the utmost importance that all conditions of the test should remain constant, especially the boiler pressure and the load. All instruments used in the test should be tested themselves before being used, in order to determine the effect of any errors to which they may be subject.

Thermometers. All important temperatures, such as feed water, injection water, condensed steam, etc., must be taken by reliable thermometers, the errors of which have been previously determined and allowed for. Good thermometers sold by reliable dealers are usually satisfactory. Thermometers with detachable scales are subject to serious errors, and should be used only for very crude work. Cheap thermometers are of little value in an engine test.

Indicators. The most important and least satisfactory instrument used in the test is the indicator. It is subject to an error of 2 per cent or 3 per cent at low speed, and this may easily be two or three times as much at high speed. It does not work satisfactorily at more than 400 revolutions per minute. If the indicator is carefully tested under conditions similar to those on the engine, the errors may be reduced to a minimum, but there will always be some uncertainty. The principal errors to which the indicator is subject have been mentioned in the instruction paper on "Steam Engine Indicators." It may, however, be well to add that for accurate work we should always use two indicators, as the long piping and joints necessary for only one causes a considerable loss of pressure and much condensation. For marine work it is customary to use only one indicator, with a three-way cock, the lower end of the cylinder usually being inaccessible.

Scales. Weighing should be done on standard platform scales. The water may be weighed in barrels provided with large drain valves which will allow the water to run out quickly. It is seldom possible to drain barrels completely, and so it is best to let out what will run freely, then shut the valve and weigh the barrel. This we call "empty" weight, and deducted from the weight "full," evidently gives us the true weight of water.

If not convenient to weigh the water, it may be measured in tanks or receptacles of known capacity, and the temperature taken, allowing the proper weight per cubic foot for water at that temperature; or it may be determined by meters.

Meters. Water meters are of two kinds, those that record the amount of water by displacement of a piston, and those in which the flow is recorded by means of a rotating disc. Piston water meters can be made very accurate, and if working under fair conditions of service they may be relied upon to a close degree. The chief error in a meter arises from the air that may be in the water. To reduce this error to a minimum, the meter should be vented, to allow the air to escape without passing through the meter. Rotary meters are good enough for very rough work, but are seldom sufficiently accurate for a careful engine test. So far as possible, weirs should not be used in engine work. They may be fairly accurate under certain conditions,

LOG OF TEST.

Time of Observation.	Revolutions.	M. E. P.	Pressures.				Temperatures.			Weights.		
			Boiler.	1st Receiver.	2d Receiver.	Condenser.	Feed Water.	Injection Water.	Condenser Discharge.	Coal.	Injection Water.	Discharge Water.

but a very little oil in the water may affect them seriously. They may sometimes be used to measure the discharge from a jet condenser, for then the volume is so large that the actual error is proportionately small.

Gages. Pressures should be measured on good gages that have been recently tested by comparison with a mercury column. The atmospheric pressure should be read from the barometer, and for accurate work this pressure should be used. For ordinary work, 30 inches, or 14.7 pounds, will do.

Calorimeter. When using superheated steam it is sufficient to take the temperature and pressure in the steam pipe, but if saturated steam is used, we must determine the amount of moisture it contains. This is done by means of a calorimeter such as described in "Boiler Accessories."

Brake. Any of the forms of friction brake described in "Steam Engine Indicators" will answer the purpose. For smooth and continuous running it is essential that the brake and its band be cooled by a continuous stream of water. The water may either circulate in the rim of the wheel or around the brake band, but it must not come in contact with the rubbing surfaces.

If the load is steady, seven or eight observations at equal intervals will usually be sufficient. If possible, the cards should be taken simultaneously, and then all the data averaged for the final result. If the load fluctuates, the cards must be taken oftener, and a greater number of observations will be required. The greatest care and accuracy must be used in all this work. In conducting a test, a careful log should be kept of the data; the outline given on page 56 being a suggestion.

THE STEAM TURBINE.

A description of the steam turbine and the general principles of the engine were given in Part I of "The Steam Engine." Now we shall discuss its efficiency. The turbine is such a comparatively new engine that there have been but few tests made, and consequently we know very much less about its possibilities than is the case with the ordinary reciprocating engine.

Probably the greatest loss in the reciprocating engine is that due to condensation and the subsequent re-evaporation at exhaust.

The exhaust cools the cylinder, so that the incoming steam meets cool walls, while in the turbine no such conditions exist. The admission takes place at one end of the engine and the exhaust occurs at the other end. The temperature gradually falls from admission to exhaust and the expanded steam never comes in contact with the part in which the higher-pressure steam works. Thus Watt's principle seems to be fulfilled, namely, "The cylinder should always be as hot as the steam that enters it." Of course there is considerable loss from radiation, but there should be very much less condensation than in the reciprocating type.

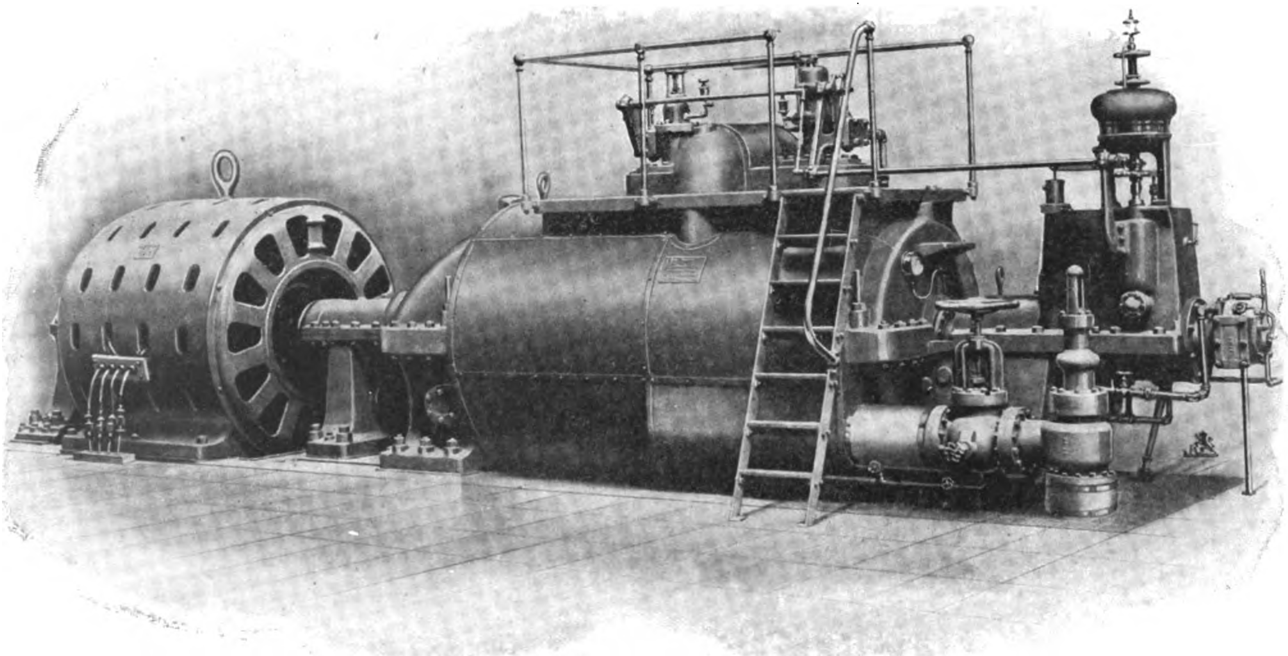
Superheating may be made use of with considerable gain in economy. There are no rubbing surfaces, no lubricants to decompose and no glands to burn out, as is the case in the reciprocating engine. The steam may be superheated to 60° or more to advantage. There being no internal lubrication, there is of course no oil to get into the condenser, and so the feed water may be used without fear of getting grease into the boiler.

Another advantage seems to be in the more complete expansion of the steam. There is little gain in the reciprocating engine by expanding the steam beyond a certain limit, because of the increased condensation. The boiler pressure cannot be increased indefinitely, neither can the expansion be carried out to the limit.

From these considerations it would seem as if the turbine ought to show much better efficiency than the reciprocating engine, and were it not for the friction of steam against the vanes of the turbine the advantage would doubtless be in its favor. Tests have shown a consumption of about 16 pounds of steam per B. H. P. per hour. Assuming an efficiency of 85 per cent, this would give about 14 pounds per I. H. P. per hour. Tests of the best modern triple expansion pumping engines have shown a steam consumption of a little over 11 pounds per I. H. P., and numerous tests of ordinary triple expansion engines have been made which show a consumption of 12 to 13 pounds.

The most recent test of which we have accurate knowledge was made on a Westinghouse-Parsons engine of 600 H. P. when running at 3,600 revolutions per minute. With ordinary steam having 3 per cent of priming, the turbine used 15.5 pounds of steam per B. H. P. per hour with a vacuum of 25$\frac{7}{8}$ inches in the

condenser. With 30° of superheating and $26\frac{1}{2}$ inches of vacuum, the steam consumption fell to 14.2 pounds. Assuming 85 per cent mechanical efficiency as before would give a relative steam consumption of 13.17 pounds per I. H.P. in the first case, and 12.07 pounds in the second case. It will be instructive to compare these figures with those given for reciprocating engines on page 53.



WESTINGHOUSE-PARSONS TURBINE-GENERATOR UNIT OF 1,000-KILOWATT RATED CAPACITY AS IT APPEARS WHEN INSTALLED

THE STEAM TURBINE

The principle of the Curtis turbine differs from that of any other type in that it permits the use of moderate rotative speeds and very compact and simple mechanism. The turbine is divided into stages, each of which contains one, two, or more, revolving

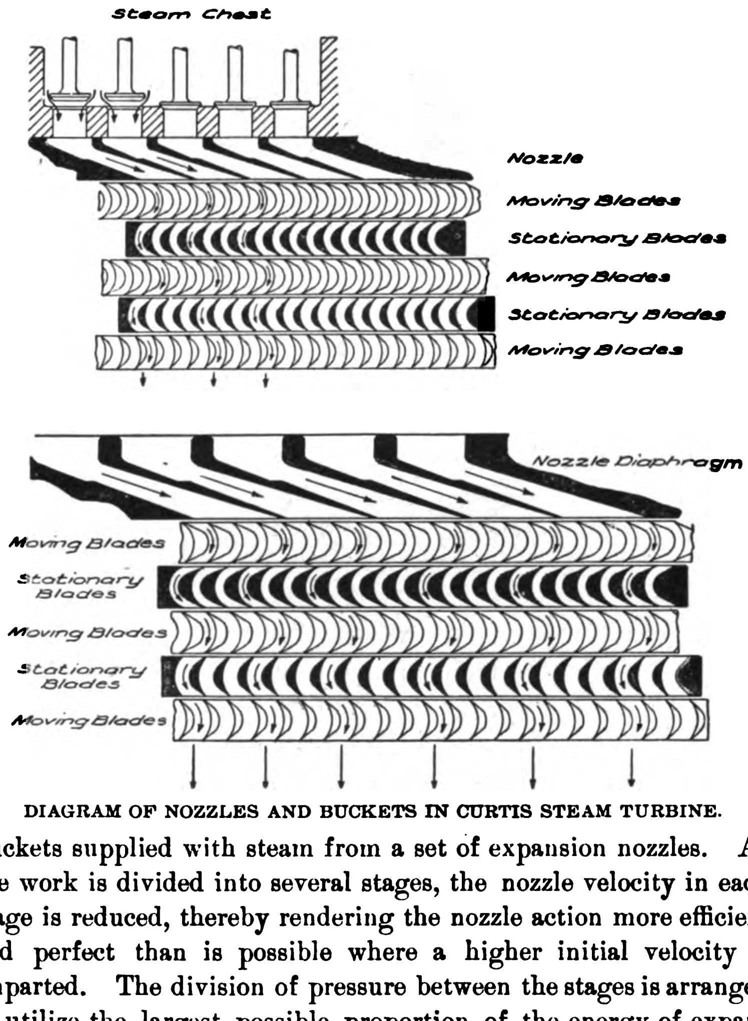
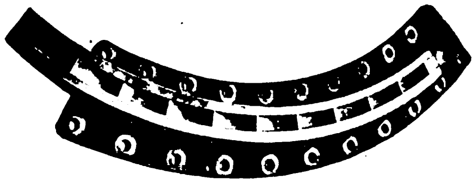


DIAGRAM OF NOZZLES AND BUCKETS IN CURTIS STEAM TURBINE.

buckets supplied with steam from a set of expansion nozzles. As the work is divided into several stages, the nozzle velocity in each stage is reduced, thereby rendering the nozzle action more efficient and perfect than is possible where a higher initial velocity is imparted. The division of pressure between the stages is arranged to utilize the largest possible proportion of the energy of expansion. The position of the moving and stationary buckets with relation to the nozzle is shown in the accompanying diagram.

Vertical Type. For turbines of large capacity the General Electric Company has applied these principles to a turbine with a vertical shaft, to avoid all imposition of weight on cylindrical bearings and tendency to shaft deflection as well as all difficulties due to irregularity of expansion or imperfections of support. The turbine is compact and of the greatest mechanical simplicity.

Step-Bearing. The step-bearing at the end of the vertical shaft supports the revolving part and maintains the revolving and stationary elements in exact relation. It consists of two cylindrical, cast-iron plates bearing upon each other and with a central recess to receive the lubricating fluid, which is forced in by pumps with a pressure sufficient to sustain the weight of the revolving part. It is apparent that the entire weight of the machine is thus carried upon a film of lubricating fluid and that there is no appreciable friction. When the flow of liquid is interrupted, the bearing is slowly worn away, but experience has shown that interruptions in the flow seldom cause any deterioration which prevents the continuance of the machine in service after the flow is re-established. The tendency of the bearing in such cases is to wear itself to a new surface so that it operates normally.

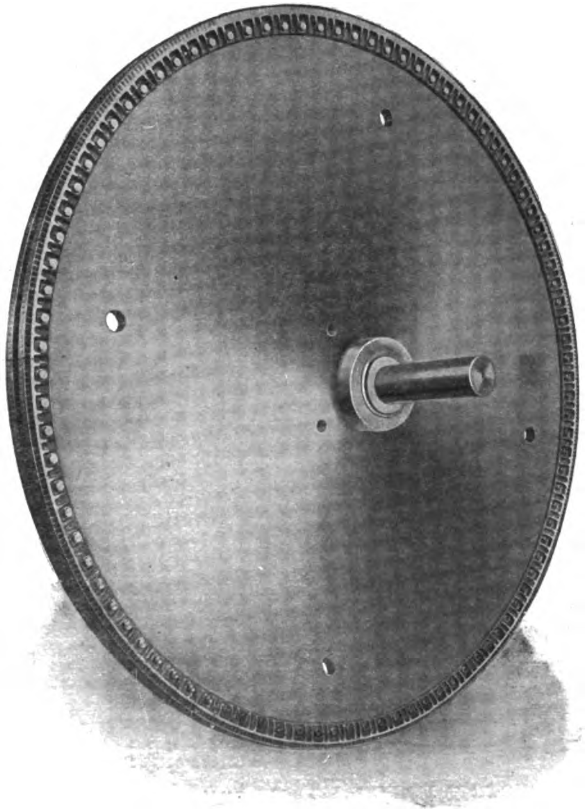


NOZZLE.

All large steam turbines are necessarily dependent upon forced lubrication. Failure of lubrication in a horizontal turbine is liable to cause serious trouble through cutting of the shaft or interference with the alignment. In the Curtis vertical type the possibility of any trouble is minimized and the simple cast-iron blocks can readily be replaced at trifling expense. In large stations, where several turbines are operated, it is desirable to install weighted accumulators which will maintain a constant pressure and also act as a reserve.

Clearances. In consequence of the exact relation maintained between the revolving and stationary elements by the step-bearing, it is possible to operate the turbines with very small clearances between the moving and stationary buckets. Experience, how-

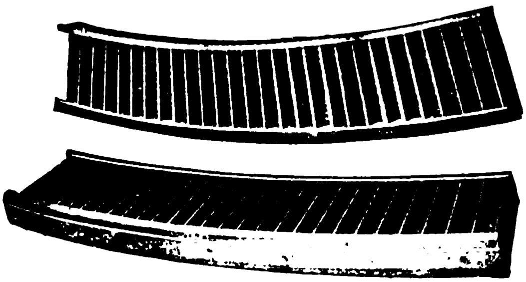
ever, has shown that the reduction of clearance beyond a certain point, is not beneficial, and that clearances less than those which are desirable for economical reasons can be used without mechanical difficulty.



REVOLVING WHEEL FOR 2000 KW. CURTIS STEAM TURBINE.

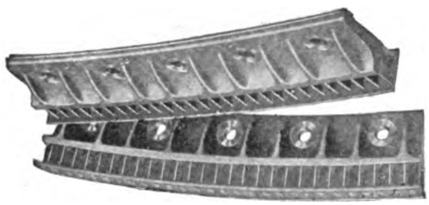
Balance. A most important matter in all steam turbine work is the balance; and the importance of good balance applies as well to vertical turbines as those that are operated in a horizontal position. When the balance is good, the bearings on the vertical turbine shaft are practically free from strain or friction. It is possible to operate these machines successfully with a considerable imperfection of balance, but a perfect balance is practicable and should be attained in every case.

Governing. The speed of these turbines with variable load is controlled by the automatic opening and closing of the original admission nozzle sections. A centrifugal governor, attached to the top of the shaft, imparts motion to levers which in turn work the



STATIONARY BUCKETS FOR CURTIS STEAM TURBINE.

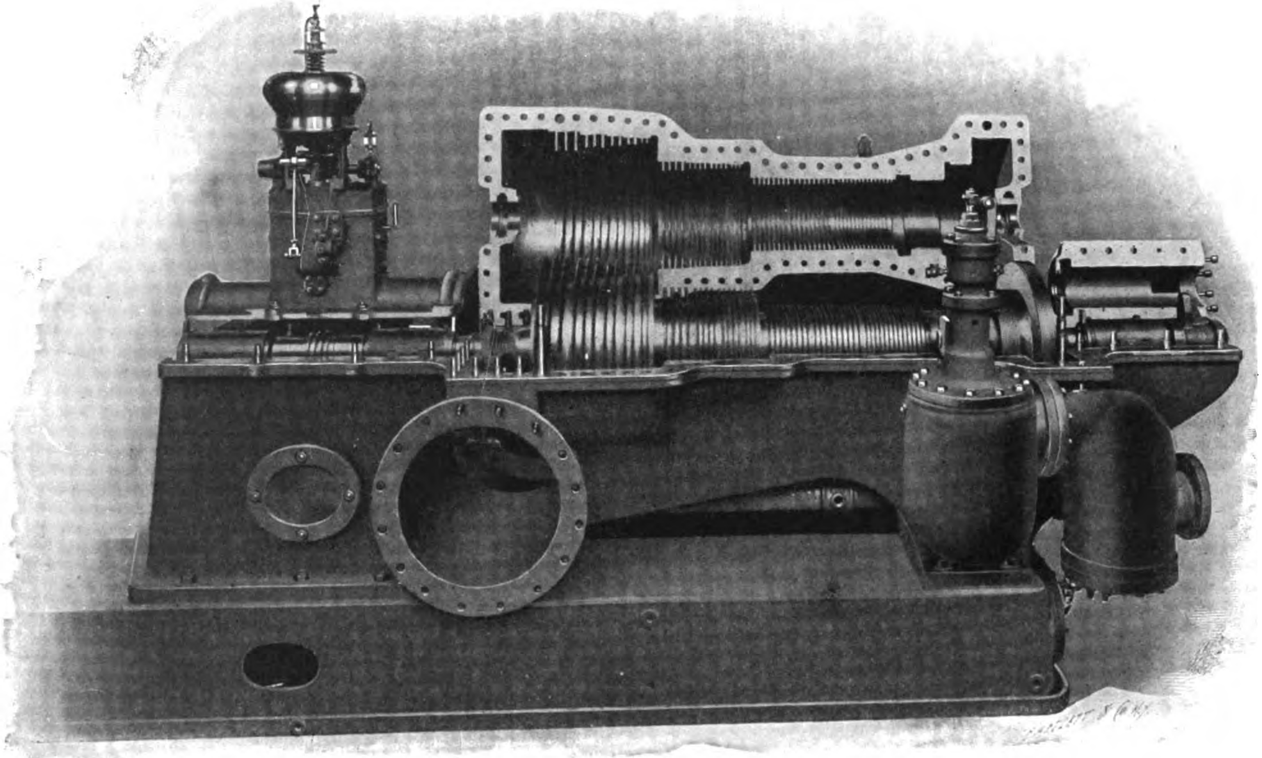
valve mechanism. There are several valves, each communicating with a single nozzle section, or in some cases two or more nozzle sections. These valves are connected to long pistons, by which the valve can be opened or closed by steam. The motion of each of these pistons is controlled by a small pilot valve which is worked by the governor mechanism. The movement of the governor mechanism moves the pilot valves successively and the main valves are opened or closed by the steam. By suitable adjustment, almost any degree of accuracy in speed control is obtainable.



REVOLVING BUCKETS FOR CURTIS STEAM TURBINE.

The steam consumption of turbines is naturally dependent upon speed, size, and other conditions, and varies in different individual designs. Machines of this type show excellent results as compared with other turbines and engines, the light-load and overload efficiency being a marked advantage. The guarantees of steam economy made by the General Electric Company are based upon results obtained with machines in operation.

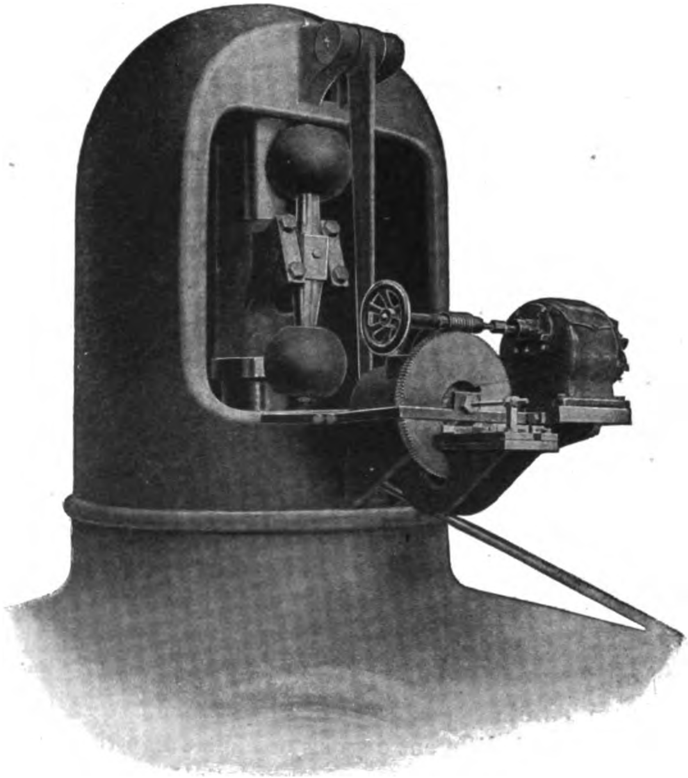
Condensers. The larger sizes of Curtis turbines are designed to operate condensing, but they are all adapted to operate non-



600 HORSE-POWER TURBINE OPEN FOR INSPECTION.

condensing, and when thus operated will carry full rated load. These turbines are designed to utilize the expansion of the steam to a high degree of vacuum, and the use of good condensing facilities is therefore desirable.

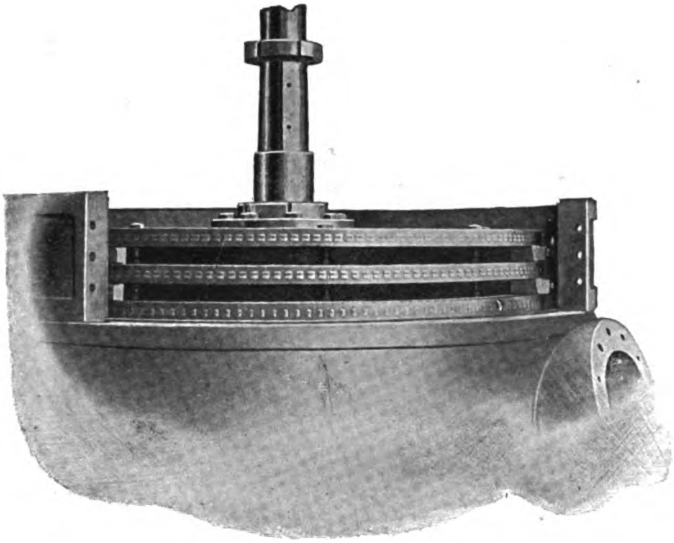
When surface condensers are used, the condensed water can be returned directly to the boilers, as it is entirely free from oil.



GOVERNOR FOR 5000 KW. TURBINE.

Experience shows that distilled water free from air has no bad effect on boilers. The possibility of so returning water is of the greatest practical value, since deterioration of boilers and inefficiency, through dirt and scale, are serious sources of expense in many power stations. In some types of steam turbines, oil is introduced in connection with balancing pistons or steam packing, and therefore this great advantage cannot be realized.

Pressure and Superheat. In these turbines steam is expanded to a considerable degree before it reaches the first buckets. High temperature in the steam is therefore not a source of practical difficulty, and steam of very high pressure and high degree of super-



500 KW CURTIS STEAM TURBINE IN COURSE OF CONSTRUCTION.

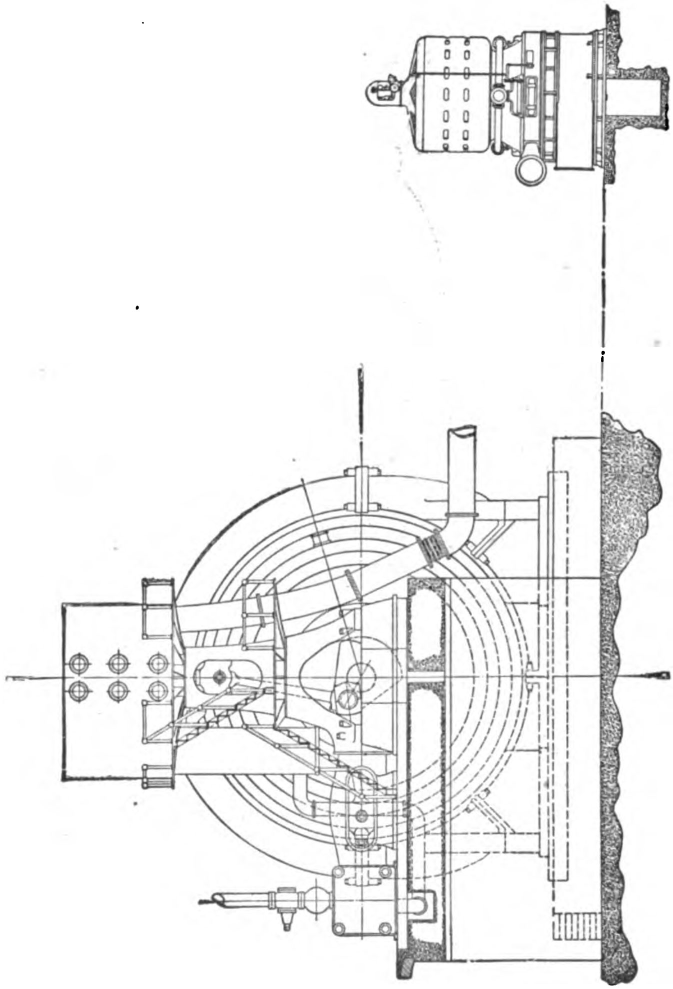
heat can be used. The reduction in steam consumption by superheat or increased pressure is as great in the Curtis turbine as in any form of steam engine.

Wear on Buckets. The question is sometimes raised as to the rate of deterioration through erosion of the buckets in the Curtis Steam Turbine. Experiments show conclusively that while the wear varies with different degrees of moisture in the steam and with different degrees of steam pressure, it is in any event negligible from the standpoint of maintenance. All buckets in the Curtis turbine can be renewed without difficulty and at small expense. In the lower pressure stages, where the density of the steam is less, no wear has been perceptible.

Applications. The speeds adopted for the Curtis turbines are such as to give the best results in the design of the generators; consequently the General Electric generators designed for operation with turbines have high efficiency, and are so proportioned that

they will carry heavy overloads without injurious heating. These turbines are built in sizes ranging from $1\frac{1}{2}$ Kw. to 5,000 Kw.

The Curtis turbine is also suitable for driving centrifugal

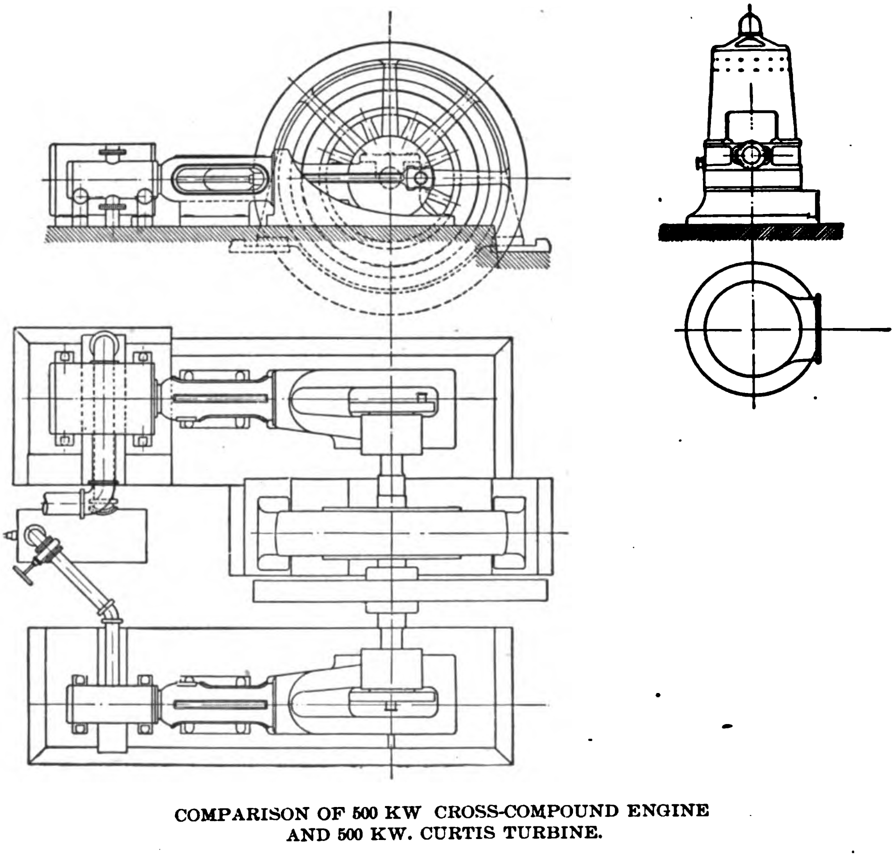


COMPARISON OF 5000 KW. CORLISS ENGINE AND 5000 KW. CURTIS TURBINE.

pumps, blowers, fans, and other similar apparatus. Turbines for such applications are being rapidly developed. In order to meet the demand for small direct-current turbines to be used for exciters, train lighting, and isolated lighting, a complete line of non-condensing horizontal shaft turbines has been developed, ranging

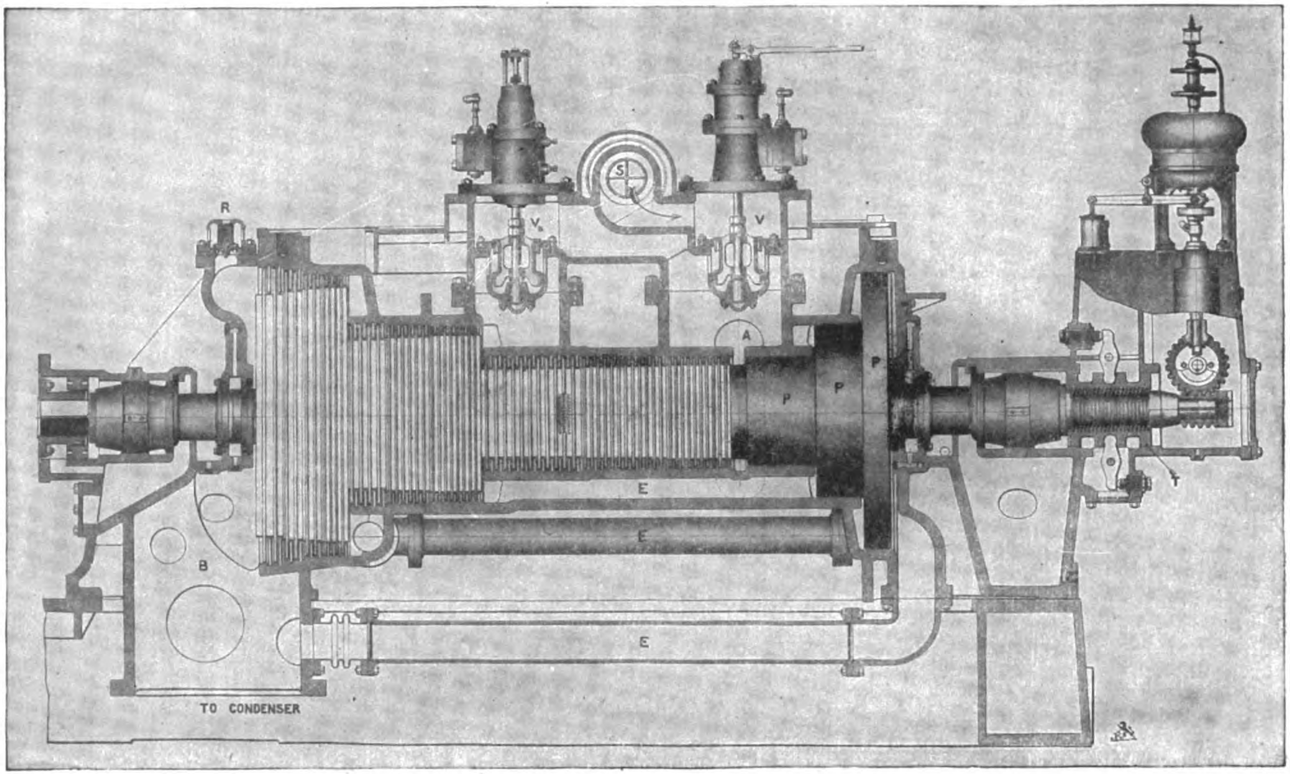
in capacity from $1\frac{1}{2}$ Kw. to 300 Kw. These machines are designed to operate at low shaft speed without the use of gearing and show relatively high steam economy when operating non-condensing. They are self-contained and are automatic in regulation.

The machine shown in the accompanying diagram has been tested under a variety of conditions at Newport, and has given



COMPARISON OF 500 KW CROSS-COMPOUND ENGINE AND 500 KW. CURTIS TURBINE.

results which illustrate very well the advantage of this type. Among other tests that were made, the machine was operated on a rapidly changing railway load; the momentary variations of load amounting to about 120 Kw. In one test the average load carried with this fluctuation was 250 Kw., and the steam consumption was 24.4 pounds per Kw. hour output, with saturated steam. Another



SECTIONAL ELEVATION OF A TYPICAL WESTINGHOUSE-PARSONS STEAM TURBINE.

test was one with similar fluctuations and with an average of 421 Kw. The steam consumption under this condition was 20.7 pounds of saturated steam per Kw. hour output. The best reciprocating engine under conditions of the first test would probably consume from 28 to 30 pounds per Kw. hour.

The Westinghouse-Parsons Steam Turbine operates on the principle that steam expanding through a definite range of temperature and pressure exerts the same energy whether it issues from a suitable orifice or expands against a receding piston.

Two transformations of energy take place in the steam turbine; first, from thermal to kinetic energy; second, from kinetic energy to useful work. The latter alone presents an analogy to the hydraulic turbine. The radical difference between the two turbines lies in the low density of steam as compared with water, and the wide variation of its volume under varying temperatures and pressures.

A typical Westinghouse-Parsons turbine is shown in section in the illustration.

The steam volume progressively increases from inlet A to exhaust B in the annular space between the rotating spindle and the cylinder walls. The entire expansion, which is approximately adiabatic * is carried out within this annular compartment which essentially resembles a simple divergent steam nozzle. There is this difference, however, that whereas in a nozzle the heat energy of the working steam is expanded upon itself in producing high velocities of efflux, in the Westinghouse-Parsons turbine the total energy, due to expansion between pressure extremes, is subdivided into a number of steps. In each step the dynamic relationship of jet and vane is such as to secure a comparatively low average velocity from inlet to exhaust, this generally varying from 150 feet per second as a minimum at the high pressure end to about 600 feet per second as a maximum at the low pressure end.

The result is of the utmost importance. With high steam velocities, excessive surface speeds are encountered, causing serious losses from fluid friction and rapid deterioration of parts from erosion. With low steam velocities, commercial speeds are readily obtained, friction loss is greatly reduced, and the depreciation of

* *i.e.*, No heat is taken in or given out by the steam cycle.

the turbine from practically the only source of wear becomes inappreciable.

The expansion of steam at any one element is typical of its working throughout the turbine. Each element consists of a ring of stationary and a ring of moving blades. The former give direction and velocity to the steam; while the latter immediately convert the energy of velocity into useful torque. The total torque upon the shaft is due both to impulse of steam entering the moving blades and to reaction as it leaves them.

A condensing steam turbine in operation affords a striking example of the conversion of heat into work. The temperature of the cylinder falls, within a distance of three or four feet, from 365 degrees Fahrenheit at the high pressure end to 115 degrees at the exhaust end, when working with 150 pounds of steam pressure and 27 inches vacuum. These temperatures remain constant during operation. There is no alternate condensation and re-evaporation as in the piston engine.

Construction. The Westinghouse-Parsons turbine in effect consists of but two essential elements—a casing, stator or stationary part, and a rotor or rotating part. A brief detailed description follows:

Rotating Element. The rotating element is built up of cast-steel drums carrying rows of blades or vanes, these being mounted on a steel shaft. These drums are arranged in three steps of increasing diameters, but the selection of three diameters is merely for mechanical convenience. Provision for the proper expansion of the steam might be made whether there be one or several diameters. If, however, a speed and diameter of rotor be selected that would permit of a convenient size of blades at the outlet, those at the inlet would become inconveniently small, and *vice versa.* By varying the drum diameters at several convenient points, the proper velocity relations between steam and vane may be preserved, and at the same time the number of different sizes of blades may be reduced to a minimum.

Opposed to the three sets of blades the spindle also carries three rotating balance pistons P, each of such diameter as to exactly balance, by means of the passages E, the axial thrust of the steam against its corresponding drum of blades. These balance pistons

revolve within the cylinder with a close fit, but are not in mechanical contact. The adjacent surfaces are provided with frictionless packing rings which offer so devious a path for the steam as to make leakage past them inappreciable. The shaft also carries a small thrust, or, more properly, adjustment bearing T, whose sole function is to maintain the normal mechanical clearances between the rotating and stationary blades. These clearances may be conveniently large without lowering the efficiency. In actual practice they are never less than one-eighth inch, and in large blades are as much as one inch.

Casing or Stationary Element. The interior proportions of the casing conform to the several diameters of the rotor and its parts. Around its inner surface are fixed rings of blades which alternate in position with the rings of revolving blades upon the rotor, and are of reverse pitch. The cylinder is divided along a horizontal plane so that by simply lifting the cover all the working parts are exposed to view.

Blades. The precise curvature and arrangement of the blades is the result of both theory and exhaustive experiment. The blades are so assembled as to admit of great ease of repair, and by a calking process which holds them so firmly to the body of the rotor that they will pull in two before they can be drawn out by force.

Glands. Frictionless glands are provided at the ends of the casing or stator to prevent the escape of steam or the influx of air into the turbine at the point of entry of the shaft. Air leakage is particularly detrimental in cases where it is desirable to maintain a high vacuum. The water-sealed glands used in the Westinghouse-Parsons turbine effectively prevent this leakage, and, further, require no lubrication. It is impossible for any oil from the bearings or the lubricating system to find its way into the steam spaces. There are no rubbing surfaces in these glands, and experience has demonstrated that wear is negligible. The water used for sealing them is small in quantity and is not wasted, as, after serving its purpose, it may be returned to the feed-water system.

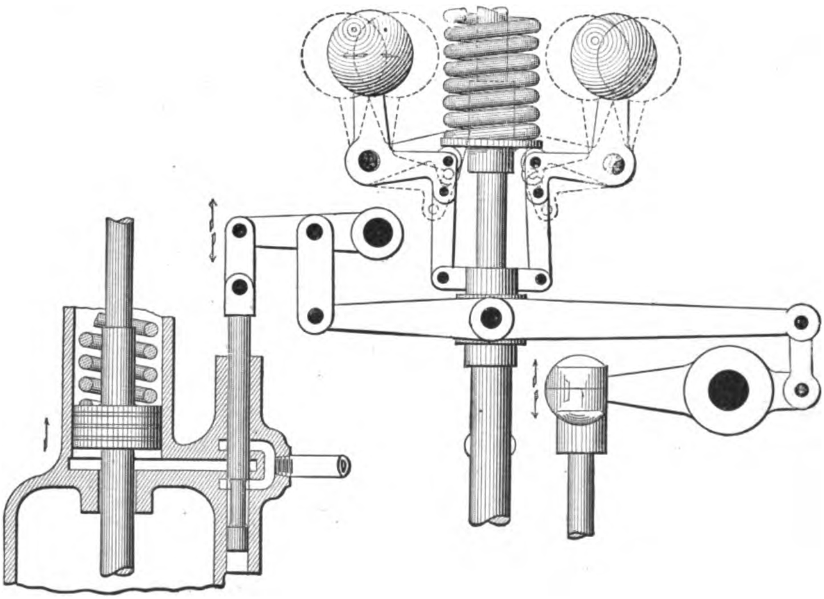
Bearings. In turbines of moderate size and therefore of relatively high rotative speeds, flexible bearings are employed in

order to permit the spindle to revolve upon its gravity instead of its geometric axis. This expedient is desirable to absorb the vibration which occurs while the turbine is passing its critical speed. The bearing consists of a nest of loosely fitting concentric bronze sleeves with sufficient clearance between them to insure the formation of oil films. These form cushions, permitting a certain amount of vibration of the shaft, but at the same time absorbing and restraining it within narrow limits. In the larger sizes of turbines, however, and in fact for all machines running below 1,200 revolutions per minute, the flexible bearing is not necessary. Instead, a split self-aligning bearing, lined with anti-friction metal, is used as in ordinary forms of moderate speed machinery.

Lubrication. In the Westinghouse-Parsons turbine the bearing surfaces are so liberally proportioned that the entire weight of the rotating element is supported upon a fluid film of oil through capillary action alone, and without the use of oil under high pressures. A small pump driven from a worm gear upon the shaft circulates oil through a closed lubricating system, comprising in the order of their arrangement—pump, oil cooler, bearings and reservoir. The oil is always supplied to the bearings at the point of least pressure; that is, at the top of the shell, from which it is distributed around the shaft. The pressure upon the fluid films is due simply to a static head of one to three feet of oil sufficient to insure thorough flushing of the bearings. It is probable that the shaft never comes in actual contact with the bearings but is separated by the oil film, as is evidenced by the preservation of the original tool marks upon the interior of the shell after years of use.

Governor. Steam enters the turbine in puffs, not in a continuous blast. Speed regulation is, therefore, accomplished by proportioning the duration of the puffs to the load. This is done by means of a small pilot valve actuated directly by the governor and which controls the steam supply through the main poppet admission valve. When the turbine is in operation the main poppet valve is continually opening and closing at uniform intervals, but the periods during which the valve is allowed to remain open are proportioned to the load on the turbine. At light load

the valve opens for a very short period and remains closed during the greater part of the interval. As the load increases the period lengthens, until finally, at about full load, the valve does not reach its seat at all and continuous pressure is obtained in the high pressure end of the turbine. On the load becoming further increased an auxiliary or secondary valve begins to open and to admit steam to the annular space at the beginning of the intermediate drum of the rotor where the working steam areas are



DIAGRAMMATIC ARRANGEMENT OF GOVERNOR MECHANISM.

greater. This increases in proportion the total power of the turbine. The operation of this secondary poppet valve is the same as that of the main admission valve, so that the governor automatically controls the power and speed of the turbine from no load to such overloads as are usually beyond the limits of generating apparatus built on normal ratings. The turbine also operates at its *best economy* at or near full rated load, although possessing at the same time large overload capacity with remarkably high efficiency.

The governor is of the fly-ball type, the ball levers being mounted on knife edges instead of pins to secure sensitiveness.

The speed of the turbine while running may be varied within the limits of the governor spring by grasping the knurled hand wheel at the top, when the spring and tension nuts may be brought to rest. Adjustment of the spring tension may then be made. This feature is particularly useful for synchronizing the speed of alternating current generators operating in parallel and for distributing the load between them when so operated. The figure shown illustrates diagrammatically the connection between the governor and the pilot valve. Variations in the speed change the height of the fulcrum of the lever on the governor spindle, which in turn varies the throw of the pilot valve relatively to the valve port. This controls the main valve and steam admission as above stated. Reciprocating motion necessary to operate the mechanism originates in an eccentric driven by the turbine from a worm on the main shaft.

Speed Limit. On the larger size turbines an automatic centrifugal speed limit governor is provided which instantly shuts off the steam supply if a predetermined excess of speed above normal should be reached.

Coupling. A flexible sleeve coupling connects the turbine to its generator. Either machine may thus be dismantled without disturbing the remaining adjustments.

The De Laval Steam Turbine, which dates back to 1882, and lays claim to be the pioneer of practical steam turbines, was first constructed by Dr. Gustaf de Laval, of Stockholm, Sweden. His first turbine was of the reaction type and was used in conjunction with his world-famous Cream Separator. In 1888 he built a turbine of the single impulse, free jet type, which differs only in improvement from the engines of the present time. The De Laval turbine has only one set of vanes, and one set of expanding nozzles, in which the complete expansion of the steam takes place in one operation, resulting in a high velocity jet.

The kinetic energy of the jet is successfully utilized by using high vane speeds, easily attained by mounting the turbine wheel, which is of special design, to withstand the centrifugal strains developed at that speed, on a flexible shaft, *i.e.*, a shaft of such dimensions as will allow of the wheel rotating about its center of mass, in place of its center of sphere. This, the "critical speed", usually takes place at about one-sixth of the rated speed. The

object attained is that all vibrations due to unequal balancing disappear on reaching this point. It can be noted by anyone who is starting up a De Laval turbine, by the sudden quieting which takes place.

The velocity of the steam jet being given, and with the nozzles at an angle of 20 degrees to the plane of the buckets, the velocity of the turbine wheel should be 47% of the velocity of the steam. The absolute velocity of the steam leaving the buckets is then 34% of the initial velocity, and the formula $\frac{29 \times 550 \times 3{,}600}{V^2 - V_3^2}$ where V and V_3 are, respectively, the initial and final velocities, will give the theoretical steam consumption. With an initial velocity of 4,000 ft. per sec. this would be 9.1 lbs. per horse-power.

Practical considerations, however, confine the De Laval wheel to a lower peripheral speed, *i.e.*, about 1,380 ft. per sec., in place of 1,880, as it should be in the case just given. This gives a theoretical consumption of 9.8 in place of 9.1.

The buckets are separately drop-forged, and are fitted into the wheel by means of a bulb shank fitting into a corresponding slot milled in the wheel.

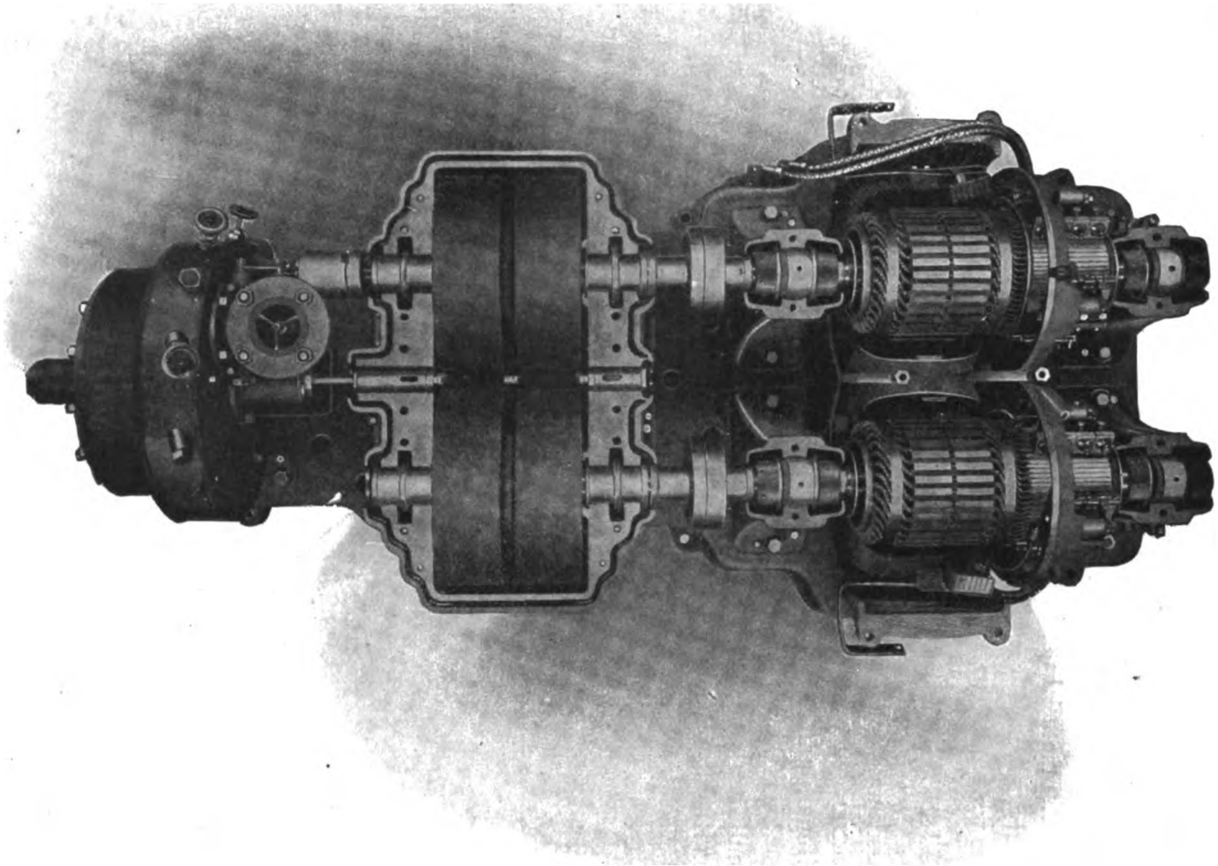
The steam nozzles are of bronze, except where high superheat is used, when nickel steel is substituted. The nozzle section naturally varies with the steaming conditions, *i.e.*, high or low vacuum, and high or low steam pressures, a greater divergence being allowed for the high pressures and high vacuums.

By closing some of these nozzles on a steady light load, the throttling effect of the steam is so decreased as to give excellent efficiencies on light loads.

The gearing is of helical design, as is seen in the accompanying figure, which shows a 110 H.P. turbine dynamo with upper half of gear case and field frame removed.

The high speed shaft bearings are all lined with special high speed metal, and are oiled by gravitation feed, the oil being used over and over in continuous cycle. The low speed bearings are oiled automatically by rings.

The speed governing is performed by a centrifugal governor placed on a low speed shaft and operating on a balanced valve.



110 H.P. TURBINE DYNAMO—UPPER HALF OF GEAR CASE AND FIELD FRAME REMOVED.

In addition to this means of control, an air valve is placed on the wheel case, operated by the governor above a certain speed, admitting air into the wheel case and thereby producing a braking effect on the wheel.

The material used in the construction of the De Laval turbine is only of a high grade. The wheel case is of cast steel; the turbine wheel and shaft of forged nickel steel; the gear wheels of cast iron, with a mild steel rim put on by hydraulic pressure.

The De Laval steam turbine is used on all forms of high speed work, and is especially adaptable for direct connection to electric generators (both alternating and direct current), for centrifugal pumps, and for air or gas blowers.

The De Laval steam turbine will work under all existing conditions from 60 lbs. non-condensing up to 250 lbs. and over, condensing; the nozzles naturally being made to suit the particular conditions.

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